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A Thermodynamic Analysis of District Heating

Svend Frederiksen

Lund 1982

**A Thermodynamic
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District Heating**

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PREFACE

The present thesis is the result of my research work as a member of the staff of the Department of Heat and Power Engineering at Lund Institute of Technology, Sweden. Part of the work was sponsored by the Swedish Council for Building Research.

I wish to express my gratitude to my supervisor, Professor S. Borglin for introducing me to the subject of the thesis and for his support in the various phases of the work. I am also indebted to Professor L. Thörnqvist and to Dr T. Torisson of the Department for inspiration and for valuable criticism.

In the course of the work I have had the opportunity to consult specialists on various topics related to the thesis. I am particularly grateful to the following persons: Dr J. Claesson of the Department of Theoretical Physics and Mechanics in Lund. Professor E. Frederiksen (my father) of the Royal Technical University in Copenhagen. Dr S. Lindahl, formerly of the Department of Automatic Control in Lund, presently at the South Swedish Power Board (Sydkraft AB). Professor H. Munser of the Technische Universität in Dresden. Professor W. Traupel of the Eidgenössische Technische Hochschule in Zürich.

Since the work was carried out within the academic world, it is not based on first-hand experience from practice. However, discussions with a number of persons active in companies and institutions operating and designing plants have been helpful to supplement the picture I have been able to establish by reading published reports. I wish here to thank these people and to express my respect for their work.

I am also grateful to a number of persons who in various ways assisted me in the process of completing the manuscript. Mrs B. Lindberg, Lecturer at the Department of English at the Lund University, reviewed most of the English text. Mr M. Bengtsson, MSc, converted my equations and flow-charts into computer programmes. Mrs C. Hedlund, Departmental Secretary, typed the manuscript. Mrs Y. Fernvall, BA, and Mrs S. Åkesson of the Department of Heat and Power Engineering assisted in collecting literature for the thesis.



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NOMENCLATUREEnglish Letter Symbols

A_1	heat transfer surface area of turbine condenser
A_2	heat transfer surface area of consumer's heat exchanger
A_3	heat transfer surface area of building radiator
A_{hw}	heat transfer surface area of consumer's heat exchanger for hot water services
A_r	heat transfer surface area of consumer's heat exchanger for heating of radiator system water
a	sensitivity of turbine output to variations in steam condensing temperature
b	sensitivity of turbine condenser heat delivery to variations in steam condensing temperature
C	cost function
c	specific heat for water
$c_{11}, c_{12}, c_{21}, c_{22}$	coefficients in eqs. (3-17) and (3-18)
c_e	specific cost of electric power
c_f	specific cost of heat rate delivered with fuel in steam boiler
c_{pa}	specific heat of boiler air
D, d	pipe diameter
D_x	diameter of supply pipe
D_y	diameter of return pipe
e	sensitivity of feedwater pump work to variations in steam condensing temperature
H	heating value of fuel (higher or lower values)
h	enthalpy
I_{IO}^*	dimensionless insulation number for supply heat loss
I_{IRO}^*	dimensionless insulation number for return heat loss
I_{III}^*	dimensionless insulation number for internal heat loss from supply to return pipe
i	sensitivity of building space heating to variations in ambient temperature
j	sensitivity of heat loss from supply pipe to variations in supply water temperature
k	sensitivity of heat loss from return pipe to variations in return water temperature
L	length of district heating pipes
L_1	length of turbine condenser (defined in section 4.3)
L_2	length of consumer's heat exchanger (defined in section 4.3)

l_{01}	sensitivity of internal heat flow between supply and return pipe to variations in supply water temperature
l_{10}	sensitivity of internal heat flow between supply and return pipe to variations in return water temperature
M_{ab}	relative size of primary water mass flow to heat exchanger for building space heating in extended system model
m	water mass flowrate in district heating system
m'	water mass flowrate in building radiator system
m_a	primary water flow to heat exchanger for building space heating in extended system model
m_b	primary water flow to heat exchanger for building hot water services
m_f	fuel flowrate
m_s	steam flowrate
m_{st}	maximum steam flowrate (turbine admission valve fully opened)
n	exponent in equation for radiator heating (eq. (3-9))
P_c	electric output from turbo-alternator
P_{e1}	P_e minus P_{eb}
P_{e2}	net electric output (P_{e1} minus P_{ec})
P_{ea}	power consumption of steam boiler (fans etc.)
P_{eb}	power consumption of boiler feedwater pump
P_{ec}	power consumption of district heating circulation pump
P_{ed}	power consumption of internal building radiator pumps
P_q	heat flow
P_{q1}	heat flow added to district heating system by generating plant
P_{q2}	heat delivered by district heating system to consumers
P_{q2D}	building heat load at ambient design temperature
P_{q2r}	building heat load when standard insulation is applied
P_{qIO}	heat loss from supply pipe
P_{qIIO}	heat loss from return pipe
P_{qIII}	internal heat loss from supply to return pipe
P_{qa}	steam boiler heat loss
P_{qc}	total heat loss from district heating system
P_{qhw}	heat requirement for hot water services
P_{qr}	heat requirement for radiator heating
P_{qrad}	radiation heat loss from steam boiler
P_{qstack}	stack loss from steam boiler
$P_{qunburnt}$	boiler loss due to unburnt fuel
p	pressure
p_1	exponent for change of heat transfer coefficient in turbine condenser

P_2	exponent for change of heat transfer coefficient in consumer's heat exchanger
P_{2A}	exponent for change of heat transfer coefficient on primary side in consumer's heat exchanger
P_{2B}	exponent for change of heat transfer coefficient on secondary side in consumer's heat exchanger
P_{hw}	exponent for change of heat transfer coefficient of heat exchanger for building hot water services
P_r	exponent for change of heat transfer coefficient of heat exchanger for space heating in extended system model
q'_1	heat flow per unit length of supply pipe
q'_2	heat flow per unit length of return pipe
q''_1	heat flow per surface unit of supply pipe
q''_2	heat flow per surface unit of return pipe
R	recovery coefficient referring to transfer of heat developed by pressure drop in water passing heat exchanger
R'_{10}	heat flow resistance per unit pipe length for heat loss from supply pipe
R'_{20}	heat flow resistance per unit pipe length for heat loss from return pipe
R'_{12}	heat flow resistance per unit pipe length for internal heat loss between district heating supply and return pipes
S^*_1	dimensionless heat transfer surface area of turbine condenser
S^*_2	dimensionless heat transfer surface area of consumer's heat exchanger
s	curvilinear coordinate inverse of dimensionless DH water mass flowrate m^*
t	temperature
t_1	supply water temperature of building radiator system
t_2	return water temperature of building radiator system
t_a	ambient temperature
t_{aD}	ambient design temperature for sizing building heating system
t_c	steam condensing temperature
t_h	building room temperature
t_{hw}	building hot water temperature
t_{in1}	district heating return temperature at generating plant
t_{in2}	district heating return temperature at consumer's connection
t_i	intermediate district heating water temperature between back-pressure plant and heat-only boiler in extended system model
t_{ina}	primary return water temperature of flow from heat exchanger for space heating
t_{inb}	primary return water temperature of flow from heat exchanger for building hot water services

t_{kw}	temperature of building cold water supply
t_s	soil temperature
t_{stack}	stack gas temperature
t_{out1}	district heating supply temperature at generating plant
t_{out2}	district heating supply temperature at consumer's connection
U_1	overall heat transfer coefficient of turbine condenser
U_2	overall heat transfer coefficient of consumer's heat exchanger
U_{2A}	heat transfer coefficient on primary side of consumer's heat exchanger
U_{2B}	heat transfer coefficient on secondary side of consumer's heat exchanger
U_{hw}	overall heat transfer coefficient of heat exchanger for building hot water services
U_r	overall heat transfer coefficient of heat exchanger for building radiator system of extended system model
u	specific internal energy flow velocity
x	part of district heating pumping energy dissipated in supply pipe
x_1	ratio between heat transfer coefficients on the two sides of the heat transfer surface of the turbine condenser
x_2	ratio between heat transfer coefficients on the two sides of the heat transfer surface of the consumer's heat exchanger
x_{hw}	ratio between heat transfer coefficients on the two sides of the heat transfer surface of the heat exchanger for hot water services of the extended system model
x_r	ratio between heat transfer coefficients on the two sides of the heat transfer surface of the heat exchanger for radiator heating in the extended system model
w	part of district heating pumping energy dissipated in turbine condenser
y	part of district heating pumping energy dissipated in return pipe
z	part of district heating pumping energy dissipated in consumer's heat exchanger

Greek Letter Symbols

α	ratio between power and heat delivered from back-pressure plant
η_b	boiler efficiency
η_t	efficiency of turbine expansion
θ_1	temperature difference between district heating supply temperature and radiator water supply temperature of consumer's heat exchanger

θ_2	temperature difference between district heating return temperature and radiator return temperature of consumer's heat exchanger
θ_3	difference between radiator water temperature and room temperature
θ_p	temperature difference defined by eq. (3-88), related to heat developement in heat exchanger due to water pressure drop
λ	heat conductivity
μ	ratio of mass flowrates between primary and secondary sides of consumer's heat exchanger
ρ	water density

Nomenclature Conventions

(presented in more detail in section 1.4)

index 'o'	designates that the variable is taken at reference operational mode
index 'or'	designates that the variable is taken at both reference operational mode and at reference values for system design parameters
asterix *	is added to a variable to indicate that is a dimensionless variable

1. INTRODUCTION

1.1 Focus of Interest

This work is devoted to a thermodynamic study of district heating (DH), with special emphasis on applications where heat for space heating and for hot consumption water is supplied to built-up areas from combined heat and power (CHP) plants.

As has been pointed out many times over the years and demonstrated in practice, the CHP principle offers possibilities of significant reductions in primary energy consumption in comparison with production of electricity and heat in separate plants. The CHP principle was realized in several places in the world (e.g. refs. [1-1] and [1-2]) already at the beginning of the expansion of electric power networks early in this century. Today, CHP in some countries plays an important role in the energy sector [1-3], [1-4], [1-7]. In many countries in the industrialised world CHP is seen as one of the most important ways of reducing primary energy consumption on a national level [1-5], [1-6]. It is also important to observe, however, that despite its attractions at the moment CHP contributes to no more than some few percent of the world's total production of electricity [1-7]. The limitation is partly due to the fact that DH, for obvious reasons, is uncommon in climatic regions with mild winters. But this is not the only explanation: the installation of DH networks calls for large investments and requires continuous dedication to the expansion of the networks, usually over several decades, on the part of some organizational body, in Sweden, for instance, typically a municipal authority.

Seen in a historical and international perspective, DH has been closely coupled to the CHP principle. In the Federal Republic of Germany, for instance, the country of Western Europe with the largest total length of installed DH pipelines, this coupling has by tradition been particularly strong [1-3]. Yet DH systems are sometimes developed without the intention of utilizing the CHP principle. An example of this was the installation in Denmark in the post-war period, of many relatively small DH systems on a private basis, where the incentive was, besides conveniences to property owners, possibilities of distributing cheap heat from large heat-only boilers, based on heavy fuel oil in place of the more expensive distillate oil, needed for individual domestic boilers. But at present the Danish

Government's energy plans include a strong commitment to the CHP principle, and many of these DH networks are now being integrated into larger public networks, connected to CHP extraction plants close to the sea [1-6].

In Sweden the DH sector was insignificant up to the very end of the Second World War. But in the fifties, a number of the larger towns began to develop DH schemes based on CHP plants, partly with examples from Denmark as a model [1-8], [1-9]. Those CHP plants, along with some fossil-fueled condensing plants, represented a new element in the previously completely hydro-power dominated national electric power system. In the last 30 years, DH in Sweden has expanded at a rate which, seen in an international context, has been particularly rapid [1-3]. According to present Government plans [1-10] the expansion is expected to continue, so that around 1990 about 40% of the nation's total heating supply should be served by DH systems. However, this development is foreseen to continue in a partial divorce of DH from the CHP principle. Mainly due to the commissioning these years of a number of nuclear power plants, the need for additional new plants will be small, so that only few CHP can be expected to be built in the near future. A recent consultative referendum has stipulated that over a period of years, with the year 2010 as an ultimate limit, all nuclear plants should be decommissioned from the nation's power system. Therefore it is foreseen that in the latter part of this century, the need for new power plants may once again increase dramatically. To a great extent Government plans count on fossil CHP plants to fill the possible gap in power capacity.

A relatively recent development has been the introduction of big heat pumps as sources of heat for DH systems. Still in an embryonic stage, this technology may, at least in some countries, develop into an important addition to the spectrum of DH heat production plant types [1-11], [1-12]. In the present work this variant will not be included in the analysis; however, the thermodynamic theory presented here appears to be well suited to a modification to systems including big heat pumps.

Since DH pipelines are very costly, over the years great effort has been put into the development of new pipeline technologies which are cheaper than traditional designs. Innovations include both technologies, already accepted in practice by some DH authorities, and more speculative designs. Research projects in the field have concentrated mainly on design problems. From the discussion in section 4.2 it will appear that a closer examina-

tion of technological boundaries of DH from a thermodynamic point of view may add important aspects to the question of choice of pipeline technology.

DH pipelines, often transporting heat over long distances and distributing it to large built-up areas, unavoidably give rise to losses in both heat and pressure, i.e. thermodynamic losses. Heat losses may typically amount to 10% of the heat energy transported by the system, but both considerably smaller and larger fractions have been measured [1-13], [1-14]. The pressure losses make necessary the addition of pumping energy to the DH system. Usually they amount to no more than 1% (or even less) of the transported heat energy, but in cases where the system includes long transmission lines from a production plant to the distribution network, the percentage may be several times larger. Heat losses are negative in the sense that they increase primary energy consumption for a given size of the heat supply to the buildings connected to the system. However, in cases where the production is based on a steam back-pressure cycle, the greater heat flow to the system, caused by the heat loss, makes possible a greater electric output, so that in such cases heat losses are not entirely negative. Pressure losses are not losses in the sense of the First Law of Thermodynamics, as the pv-energy dissipated with pressure drops is converted into heat within the system, i.e. in pipes, in heat exchangers, etc. However, they imply a degradation of mechanical energy, often produced from high-cost electric energy.

A recurrent problem in DH is how to select supply and return temperatures of the heat carrying medium, usually water. For a system with given dimensions, the supply temperature will normally decrease when the flowrate is increased at a given heat load. Hereby the heat losses will usually be reduced. This means that for a DH system, where the heat supply is served by a heat-only boiler, a proper supply temperature may be selected by balancing the two types of losses on some suitable economic basis.

When the heat supply for a DH system comes from, not a heat-only boiler, but a back-pressure plant, a further type of effect must be taken into consideration when selecting the water flow-rate, and thereby the supply water temperature. This is the fact that the electric output from such a power cycle depends a great deal on the level of the back-pressure, which in turn is determined by the temperature level of the DH system. A typical figure for this kind of sensitivity, expressed in relative change of out-

put versus temperature change is: $0,5\%/^{\circ}\text{C}$. This sensitivity in combination with a reduction of DH heat losses, gives an incentive to keep the supply temperature low, an incentive that must be balanced against an increase of the DH pumping power, at an increased water flow-rate.

The counteracting effects on the turbo-alternator output P_e and on DH pumping power P_{ec} of changes in supply temperature can be conveniently summarized by considering the difference between those two variables, i.e. the net electric output P_{e2} , as seen from fig. 1-1. When, for a given heat supply P_{q2} from a system with given dimensions, the supply temperature is to be selected, this may be done rationally by balancing changes in P_{e2} against changes in the heat consumption rate, $m_f H$, expressed as the product of the fuel flow and the heating value (higher or lower value, depending upon which is the appropriate one). Note that balancing P_{e2} against the heat loss, P_{qc} , would be less rational, since P_{qc} , unlike $m_f H$, cannot be assigned any direct economical value.

In this work a main theme will be to examine how the net power output P_{e2} varies with changes in the DH systems. Such changes will include both variations in system flowrate, i.e. in operational strategy for the DH systems, and variations in system dimensions, e.g. in pipe length and diameter. This will represent an attempt at a fairly systematic examination of how the net electric output can be varied and maximized by manipulating the DH system. At the bottom of fig. 1-1 is shown a typical temperature variation throughout the system. It is seen that here both the hot consumption water temperature and the water temperatures of the internal heating system of the building(s) lie roughly halfway between the room temperature and the condensing temperature of the back-pressure turbine. This indicates that in terms of a potential for improving P_{e2} , the DH system displays possibilities of the same order or magnitude as the internal heating systems. In a number of investigations it has been discussed to what extent DH water temperatures may be lowered by changing both dimensions of radiator surfaces and secondary water flow in radiators connected to DH systems [1-15], [1-16], [1-17]. It appears that in many cases in practice radiators have been designed with ample marginal, so that the DH supply temperature may be lowered from its normal level, with only slight or negligible modification of the radiator systems. Also a lowering of the hot consumption water temperature may be considered. In Sweden until now it has been standard in practice to design DH systems for maximum (at minimum

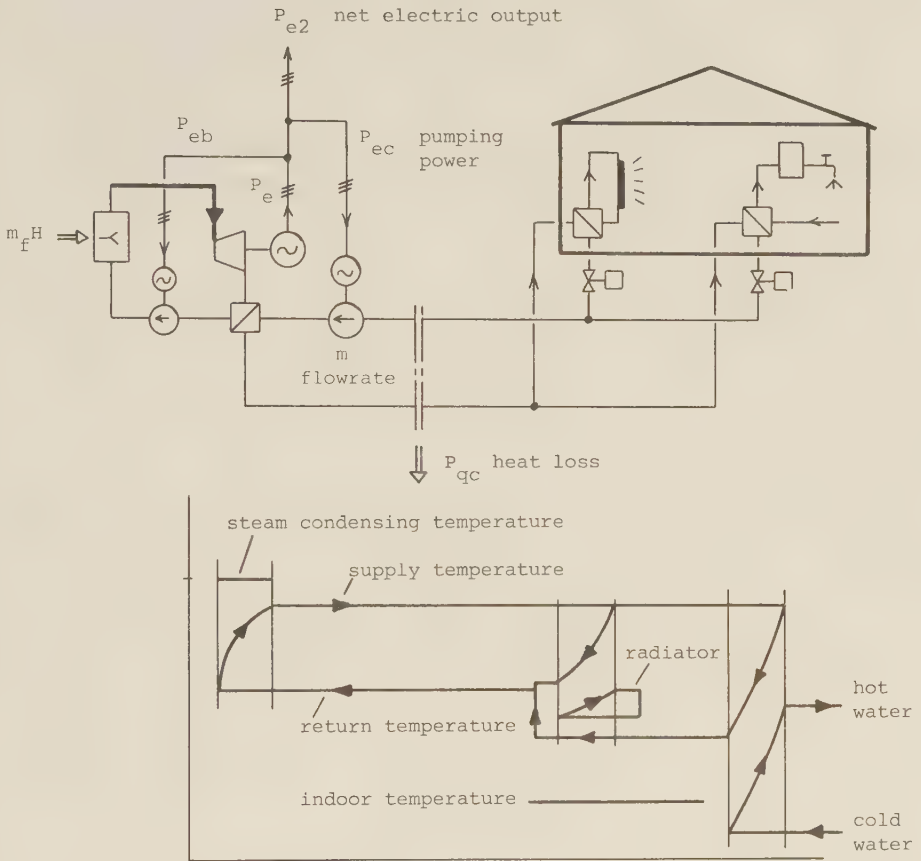


Fig. 1-1 Schematic of district heating system
 with heat supply from a steam back-
 pressure plant.

ambient temperature) supply and return temperatures of 120°C and 70°C , respectively, combined with a lowering of the supply temperature at lower heat loads. In the warm part of the year the supply temperature is held constant, usually around 75°C . Abroad both higher and lower temperature levels have been employed [1-18], [1-19]. In Germany, supply temperatures up to about 200°C have sometimes been used. In Sweden the DH water is nearly always separated from the water of the radiator systems by heat exchangers. In countries where operation without such heat exchangers is practiced, lower DH supply temperatures, sometimes even below 100°C at maximum heat load, are not infrequently applied.

Having learned that back-pressure turbines, as well as other types of production plants connected to DH systems, improve their operation with a lowering of DH temperatures, it is all too easy to push the case for low temperatures too far. It should be remembered that the operation of DH systems over a period of years is a dynamic process, where a fast expansion of the network is often a basic requirement for economy. An extremely low supply temperature will tend to increase DH installation costs, so that the network expansion may be hampered. In cases where such conditions prevail, the accumulated electric energy production of a back-pressure plant over a period of years may be lowered, not increased, by a lowering of the DH supply temperature: Although the net electric output may increase for a given heat load, this may be more than outbalanced by the negative effect of a smaller heat load, due to a slower expansion of the network. Therefore, in this work we are going to take a fairly neutral attitude towards the question of 'best' DH water temperatures. The emphasis will be on investigating consequences of various temperature choices, rather than arriving at recommendations for 'optimal' temperatures. In section 1.3 questions on the proper evaluation of system performance will be discussed more in detail.

When examining variations in net electric output and in heat consumption rate for systems like the one displayed in fig. 1-1, these variables will be shown as functions of the ambient temperature, t_a , being the main external variable determining the amount of heat required in buildings. For instance, we shall see how the DH supply temperature in system models must be varied with t_a to satisfy the criterion of maximum P_{e2} . Since the output from a back-pressure plant is closely tied to the amount of heat supplied to the DH system, the general tendency will be that P_{e2} increases

Fig. 1-2

Electricity demand in Sweden from July 1979 to June 1980 vs. a geographical mean of the ambient temperature.

Based on statistics from [1-21].

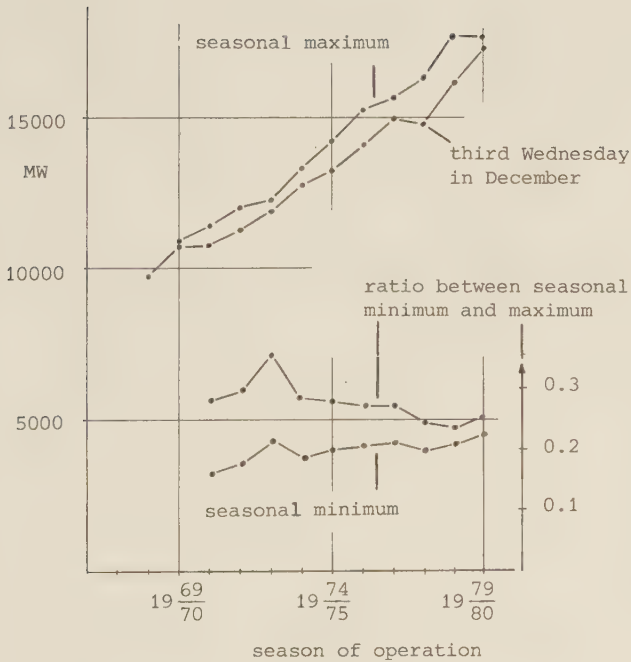
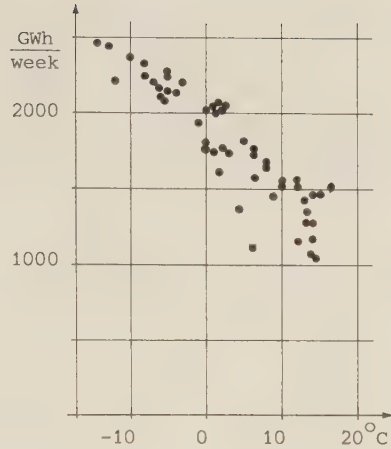


Fig. 1-3

Change in Swedish aggregate electricity production.

Based on [1-20], partly a reproduction, with permission from the author.

with a lowering of t_a . The analytical apparatus developed in following sections will enable us to quantify this type of relationship under various conditions. Of practical importance will be the examination of how the function $P_{e2}(t_a)$ is influenced by the addition of a peak heat-only boiler to a back-pressure plant. This is a very common combination; installed production capacity in back-pressure plants is usually considered too expensive to justify that the back-pressure turbine is designed big enough to cover maximum heat load which occurs only rarely. As we shall see, such an addition of a heat-only boiler tends to reverse the slope of the $P_{e2}(t_a)$ function.

The functional relationship between net electric output from back-pressure plants and changes in ambient temperature, may be compared with the statistical variation of electricity demand with ambient temperature for the big electric grids to which the plants may be connected. As an example, fig. 1-2 shows this type of relationship for Sweden during one season of operation. The week-by-week consumption of electric energy versus ambient temperature is depicted, the latter taken as a geographic mean over the country, weighed with respect to distribution of consumption. It is seen that in extreme cold the demand is around twice the demand in summer. Although there is some scattering, the correlation between demand and ambient temperature appears quite significant. Fig. 1-3 shows how year-by-year extreme loads have developed in Sweden over a ten-year period. The ratio between minimum and maximum is also shown. It can be seen that this ratio tends to decrease. Probably this reflects an increased weather sensitivity in the electric power system, which can mainly be attributed to a growth in the use of electric resistance heating for buildings. Similar trends have been observed in other big power systems [1-22]. Since the substitution of electric heating for individual oil boilers is made extensive use of at present in order to reduce the oil consumption of nations, this type of weather-dependancy has become a matter of growing concern. One type of countermeasure on the part of electric utilities has been to differentiate tariffs over the season of operation. Another type of contribution to a solution of the problem would be to design power plants for an increased output at low ambient temperature. This will be a major concern in our analysis of CHP plants.

To sum up, the aim of the present study can be formulated in the following way:

To investigate how the net electric output P_{e2} from a back-pressure plant serving a DH network varies with changes in the ambient temperature, how P_{e2} can be varied (and maximized) by different choices for DH operational mode, and how P_{e2} is affected by changes in design parameters of the DH system.

The results of this investigation should be of interest, both to engineers responsible for the design and operation of such plants, and for planners matching the operation of CHP plants with other types of generating capacity in a national power system.

1.2 Comments on Literature

The number of references of relevance to the scope of this study is large. Although many of these works are helpful in establishing a correct picture of the characteristics of various system elements, only few references demonstrate explicitly characteristics of the entire system, from the heat consumers to the CHP plant.

For many decades it has been characteristic of DH as a branch of technology that it has developed in a very pragmatic way, where much knowledge has been accumulated in operating and consulting companies without being published. In 1970 Weber [1-27] wrote:

"Eine umfassende Darstellung der Fernheizung ist m.W. im deutschsprachigen Bereich noch nicht erfolgt, wäre aber für die Fachwelt dringend erforderlich".

Meanwhile the situation has improved, due to the publication of several comprehensive books on DH [1-1], [1-28], [1-29]. Munser's textbook [1-1] is valuable, among other things in that it summarizes knowledge developed in Eastern Europe and in the Soviet Union, much of which has hardly been read in the West. Homonnay's book [1-28] also contributes to this kind of technology transfer. The handbook [1-29] mainly describes practices and experiences from design and operation. The first edition, which appeared in 1973, contained, among other things, a systematic survey of the various options for layouts of consumer connections. However, its account of operational modes for DH systems was rather summary. One figure (Bild 4.13) showed water supply and return temperatures and water flowrates as a function of the ambient temperature; while the temperature difference varied continuously, there was a jump in the variation of the flowrate, which is obviously contradictory to the First Law of Thermodynamics. The second edition, published only recently, has been revised thoroughly, and the figure mentioned no longer appears in the text.

Since Weber made his statement on DH literature, cited above, a major contribution to the documentation of information on DH has been the German journal "Fernwärme International", which has been published regularly since 1972. From the list of references (Chapter 7) it can be seen that a great deal of the works drawn upon here have been published in this journal.

Some comprehensive accounts on DH have been published in the English language. However, not very surprisingly in view of the relative insignificance of DH in Great Britain and in the USA, they are to a great extent concerned with describing continental European practices. [1-31], published in 1979, is an example of this. It may be designated as a good introduction to DH technology. But it is symptomatic of the language barriers in the field that it does not contain a single reference from "Fernwärme International", the main international journal on the subject, written in German.

Thus, although handbooks and textbooks certainly exist, DH technology still suffers from the lack of a common base of knowledge documentation in the form of a treatment which sums up practical and scientific results, and which has been widely read by people active in the field in various countries. With this state of affairs in mind, the author of the present investigation has been cautious about taking too much basic and elementary knowledge on part of the reader for granted. Therefore, the well-informed reader may find some sections, perhaps in particular section 3, more textbook-like than that which may be usual for dissertations. The ultimate goal here will of course not be to give the kind of comprehensive account of elementary knowledge of a textbook, but to arrive at the numerical results presented in chapters 4 and 5.

The present contribution to DH thermodynamics is a continuation of the work of Torisson [1-23] from 1973. A main theme here was, as in this work, the variation of the net electric output from a back-pressure turbine defined as the difference between the output from the turbo-alternator, and the pumping power needed for circulation of DH hot water. Numerical results for a very simple system model were shown in the form of duration curves for net electric output, based on an assumed heat load duration curve. Although the theoretical model analysed by Torisson was highly simplified, his investigation clearly demonstrated the importance of including in the analysis the entire thermodynamic system, from the heat consumers to the generating plant. It also showed that it is possible to derive results of practical interest on the basis of thermodynamic calculations, without involving directly economical considerations.

One of the major simplifications in Torisson's analytical system description was that the DH heat and pumping losses were assumed to be of equal size at all load conditions, in spite of the fact that the pumping power varies with

the third power of the water mass flowrate, while heating losses change much less dramatically. In ref. [1-24] an improved analytical description for DH losses was presented. One feature of this description is that it takes into account that for a given heat flow to the consumers, the pumping power added to the DH system leads to a smaller heat flow from the generating plant, thus reducing the basis for back-pressure production of electric power. This kind of effect, though a simple consequence of the First Law of Thermodynamics does not seem to be generally recognized. In the present treatment of system models, the analytical description of heat losses from [1-24] will be implemented.

In [1-23] hot water services for the buildings were not included in the simple model, and the heat generating plant was assumed to be a back-pressure turbine only; no heat peaking capacity was included. The present investigation will show how those two system complications, commonly encountered in practical systems, affect system characteristics. Another extension of Torrisson's analysis will be the discussion of technological constraints represented by DH pipelines, given in section 4.2. Finally, a number of types of graphical representations of system characteristics, in addition to those presented in [1-23], will be developed in this work.

In 1964, Halzl et al [1-26] presented a thermodynamic model for a DH system, including buildings with domestic space heating and a back-pressure steam cycle with water heating in two stages and optional peaking heat generation by throttling live steam. Equations were given for rating economically thermodynamic system characteristics. A major model limitation in this work was that the flowrate of DH water was assumed to be constant at changes in heat load. The authors called attention to the fact that at high heat loads, when full steam flowrate through the turbine has been reached, further increase in heat load deteriorates the electric output, a fact which we shall be investigating quantitatively in chapter 5.

It deserves mention that the textbook [1-25] by Rietschel and Raiss contains a short discussion of the variation of electric output from back-pressure turbines with ambient temperature for various types of DH operational strategy. Diagrams are given for the turbine output, but not for the net electric output, i.e. taking into account power for the DH pump(s).

A number of works from the University of Stuttgart in the FRG [1-32], [1-33], [1-34], [1-35] have investigated the possibilities of varying strategically the electric output from a CHP plant with time by utilizing various degrees of freedom in the system, and by controlling it with an on-line computer. Here, as elsewhere on the European continent, the price of electric power is subject to great variation during the 24-hour cycle, and during the weekly cycle, so that time-dependent variation of the electric output is of great economic value. This variation may be achieved, either by utilizing the option of condensing steam in a condenser cooled by the ambient, or by storing heat in various parts of the system. The buildings requiring heat, as well as the water in the DH pipes, represent a not insignificant heat-storage capacity. In addition, hot water storage tanks can be connected to the DH network, as has been done at the CHP station of "Pfaffenwald" of the University of Stuttgart.

A vast range of problems on DH and CHP have been investigated by researchers active at the University of Dresden in the GDR [1-36] through [1-40], including the question of selecting optimal operational strategy for DH systems. To a Western reader, the criterion of maximum "Gesellschaftliche Aufwand" that is generally used as the basis here for optimizations, may be difficult to interpret correctly. However, as will be seen in section 4.1, important results from the research in Dresden are essentially consistent with results from the present, strictly thermodynamic systems analysis.

A number of dissertations, published at various universities, deal with topics more or less related to the theme of the present investigation. Suttor [1-41] calculated economically optimal operational modes for a DH system served by two heat-only boilers at different geographical locations, utilizing the heat storage capacity of the DH lines themselves in non-stationary operation.

The works [1-42], [1-43], and [1-44] all pertain to CHP plants proper, rather than the entire system, including the DH network. Instead of a back-pressure plant they treat extraction-condensation plants, where there is the extra degree of freedom of varying the ratio of steam flow for hot condensers and cold condensers. In the present work, the choice of operational mode for the DH is instead the degree of freedom when selecting optimal operational mode. In the Finnish dissertation [1-45] an attempt is made to derive an economic optimization model, from which DH pipe diameters and other system parameters can be selected.

1.3 Method of Analysis

As has been pointed out already, a main point with the investigation made in this work will be that it addresses the entire thermodynamic system, constituted by the heat consumers, the DH system, and the generating plant. Hereby, sub-optimizations - that would result from defining more limited system boundaries - can be avoided. However, it is also clear that we shall try to embrace an extensive, and in practice highly complex, system. Therefore, if it shall be possible to achieve specific, quantitative results, it will be necessary to specialize the scope of the investigation in one way or another.

One profitable method could be to select an existing DH system from practice and specialize the investigation to that system. Instead, we shall here examine theoretical and highly simplified system models. These models are going to be presented and discussed in the next chapter. The discussion below will attempt to characterize this kind of approach from a general, methodological point of view. The reader less interested in this rather abstract reasoning may wish to proceed directly to chapter 2, or possibly even to chapter 4, where results from system performance calculations are first presented. However, it should be observed that some of the author's intentions in the selection of the specific approach adopted here will be stated below.

Our method of analysis will be distinguished by three characteristics:

- It is
- 1) a systems analysis
 - 2) a thermodynamic analysis
 - 3) a simple model analysis

The term 'systems analysis' is used here in a sense somewhat more specific than simply 'analysis of a system'. To begin with, the term 'system' designates an assembly of components which are linked together functionally in a way that enables the system to act as an entity, to perform specific tasks. Thus, a system is more than an aggregate of elements. A typical example of a system is a biological organism. Here, each organ performs specific duties, which contribute to a meaningful behaviour of the organism. In a systems analysis emphasis is placed on the study of the functional relationships between the elements of the system, and on the interaction between the system and its environment. The elements are represented by characteristics describing

their performances in the interplay, while their inner structure is considered secondary. Thus, a systems analysis concentrates on a specific level of aggregation. At a lower level the elements may themselves be seen as being composed of still smaller elements, and at a higher level of aggregation the system may itself be considered an element in interplay with other large elements. In the systems analysis of this work we sometimes take element characteristics for granted, relying on relevant information in literature. The building radiator heating system will be an example of this. In other cases, specific aspects of component behaviour, not dealt with in standard literature, will be analysed in more detail. This will be the case for heat exchangers with large primary side pressure drops.

In the last two or three decades, systems analysis has been applied to a number of subjects in many fields, including economics, sociology, and biology. There has also emerged a 'general systems analysis' which claims to be an autonomous science [1-46], [1-47]. It appears to be a matter of controversy how far the case for a systems analysis can be pushed, without degenerating into empty formalism. Another point of criticism, which has been voiced in the humanities, is that systems analysis tends to 'mechanize' relationships which can only be described adequately in verbal terms. In the natural sciences, however, this kind of objection does not appear to be relevant. In the present work, systems analysis will be applied in a pragmatic and undogmatic way, concentrating on results of engineering interest. A systematic study of the relation between the present concrete subject and general systems analysis will be left for possible future investigations.

One of the merits claimed for general systems analysis is that it facilitates a multidisciplinary analysis of complex systems by establishing a common basis of terminology. In district heating, problems of terminology certainly exist. Various parts of the system belong to different disciplines of engineering. Thus, the engineer designing consumers' connections will in general have been trained in heating and ventilating engineering, while the power station control room engineer will normally have a power engineering background. They will therefore not infrequently have limited knowledge about elements within each other's areas of responsibility.

The second characteristic of the method of analysis to be adopted, is that it is a thermodynamic analysis. The aim will be to calculate important en-

ergy flows crossing system boundaries. To this end, thermodynamic relationships, involving temperatures, pressures, enthalpies, etc, are established. The control system, necessary for the thermodynamic system to function appropriately, will not be considered part of the system to be analysed. Also we shall concentrate on static operation of the system. A brief discussion of dynamic operation is given in Chapter 6.

From the systems approach it is clear that we shall not be satisfied with a pure physical analysis in the sense of developing calculation procedures by which the system behaviour may be determined, once the system dimensions are given. Our ultimate goal is an engineering one, to explore the potential of a technology, i.e. to be able to show how the system should be designed to perform in desired ways. On the other hand, by adding the adjective 'thermodynamic' to the analysis it is stressed that our results will in principle be possible to control by experiment (although no empirical verification of the results of this work has been made yet), and that model equations will be based upon available engineering literature.

An important implication of the last point is that economical relations and criteria will not be part of the subject of investigation, and that our results will not be economic optima, although at some points economic evaluations of thermodynamic results will be discussed. In relation to practical decision situations, the aim of our analysis will be to contribute to rationality in decisions, rather than to optimality. Rationality here means the ability to foresee consequences of various decisions in design regardless of how those consequences are to be evaluated. It is an unfortunate fact that many discussions on the best choice for a design parameter in DH, e.g. supply water temperature, are confused due to vague conceptions as to thermodynamic consequences of the various options. The engineering part of the problem is underrated.

For a given thermodynamic system, an economic optimization problem may be formulated in many ways. If we are seeking the best operational mode of the DH system for given system dimensions we may assign economic values to the fuel consumption and to the net output of electricity, and minimize the sum of the two. The heat charge, paid by the heat consumers, need not be included in the optimization, since at varying operational mode the heat load may be considered a constant. If we instead want to optimize design para-

meters, functions describing variations in investment costs must be defined. For instance, when optimizing the pipeline diameters, the pipeline investment costs may be assumed to be linear functions of the diameters. Also, in this kind of calculation the heat charge need not enter into the analysis, so long as the service area for DH may be considered constant. However, to many DH utilities in Western Europe, the realistic economic problem is a more dynamic one. Although the structure of the heat charges may not be fixed quite freely, the company may have some degree of freedom in determining the level of the charges. The lower the charge, the lower will the income be for a given service area, but at the same time DH may appear more attractive to potential customers, so that a more rapid expansion of market may be achieved.

An economic optimization will have to take into consideration the structure of ownership for the technological system [1-48]. For example, both the production plant and the DH system may be within the ownership of a municipal company, and electricity is sold (and/or purchased) from a regional electricity company. In that case the exact wording of the electricity purchasing contract between the two companies will be of paramount importance for any economic optimization which the DH company may carry out. Another kind of structure of ownership is represented when the production plant is owned by the regional electricity company, and district heat is sold to a municipal DH company. If here the DH circulation pump (or pumps) is also owned by the DH company, electricity generated in the power plant and electricity consumed by the pump will influence economies of different legal subjects. In that case an economic optimization based on a rating of the net electricity output should be interpreted with caution. The results from such a calculation may be quite different from that obtained in an economic calculation with differentiated ratings for the two electric powers. Still, a calculation based on net electric output may not be irrelevant even with this type of ownership structure. A difference between the operation chosen in practice and what is found at the optimization may be an indication that the contract for purchasing electricity should be revised.

A realistic economic evaluation cannot be performed without regarding the political and legal structure of the society in which the technical system operates. In some countries the DH may be sold on a market economy basis, in other countries conditions of a planned economy may prevail. Even in Western countries with a predominant market economy, legislation regulating the DH

business may be of such a character that it creates more or less monopolies on the market for heating of buildings.

When discussing the evaluation of thermodynamic characteristics of DH systems with CHP plants, not only economic consequences of decisions should be included, but also what may be designated as 'environmental' consequences. In this concept one may include, not only physical effects, like air pollution, but also effects on the environment in a broader sense. An example of this is the impact of a DH network on the geographic distribution of settlements. DH, being a business with large capital investments, may over the years lead to a concentration of buildings around main supply lines. Such a concentration may be in conflict with other, equally legitimate demands on the distribution of settlement. However, if DH from a combined power plant is combined with electrical heating of buildings, this may increase the flexibility of the public planning, by relieving restrictions on settlement distribution.

The fact that the thermodynamic analysis, as it is applied in this work, does not lead to final recommendations for the practical operation and design of DH systems with cogeneration may seem disappointing. It may be tempting to try and develop some thermodynamic theory of a more normative character, where such recommendations could be arrived at. The concept of exergy has been proposed for such a purpose. There is no doubt that an exergy analysis, applied properly to our thermodynamic systems, may give valuable results. But to hope for a pure thermodynamic analysis to give final answers to problems at the management level, would be an overestimation of the power of the exergy concept.

The third and final main characteristic of our method of analysis as stated at the beginning of this section is that it is a simple model analysis. The term 'model' seems to have become a fashionable word these days. It is often misused to give a scientific flavour to investigations where the author is vague about the assumptions underlying his calculations. The result is frequently that it is impossible to have constructive discussions about the results of such model calculations. The model becomes an idiosyncrasy of the author.

An ideal analysis based on a calculation model may be defined as an analysis fulfilling the following three demands:

- 1) The model applied is being classified according to model typology.
- 2) The assumptions constituting the model are made explicit.
- 3) The results of model calculations are tested against reality.

Of these demands the second is indispensable, while the fulfilment of the two others is certainly desirable, but not indispensable. Below an attempt will be made to partly fulfil the first point above. Chapter 2, 'Model Formulation', together with the development and presentation of mathematical equations in following chapters, should fulfil the second point. As mentioned already, no empirical verification of calculation results is going to be presented in this work. The numerical results will, however, be discussed with reference to commonly known characteristics of DH systems in practice. Thus, only a partial fulfilment of the last of the three demands is made here. The reason is a practical one, limited time and resources. DH systems, in particular when connected to CHP plants, are extensive and complicated so that experiments with them tend to be extensive and costly. Here is an obvious point for possible future research.

In this work, two simple models will be examined. The first one is an extremely simple system with heat generation in a back-pressure turbine only, i.e. with no additional water heating boiler, and, as a further simplification, no hot water services are included. In the second model variant the two types of complication are introduced. A number of specializations and simplifications in comparison with real-world systems will be common to both the two models. Most important is that instead of a large number of heat consumers and a branched DH system, we shall in the models have only one (large) heat consumer, and only a single pair of supply and return pipes. The discussion in Chapter 2 develops these points further.

In fig. 1-4 an attempt has been made to show how the two theoretical models, for which numerical calculations are performed, are related to empirical systems. At the top of the figure we have the set of all empirical DH systems with back-pressure generation plants, including both existing systems and possible future plants. The first arrow pointing downward indicates a

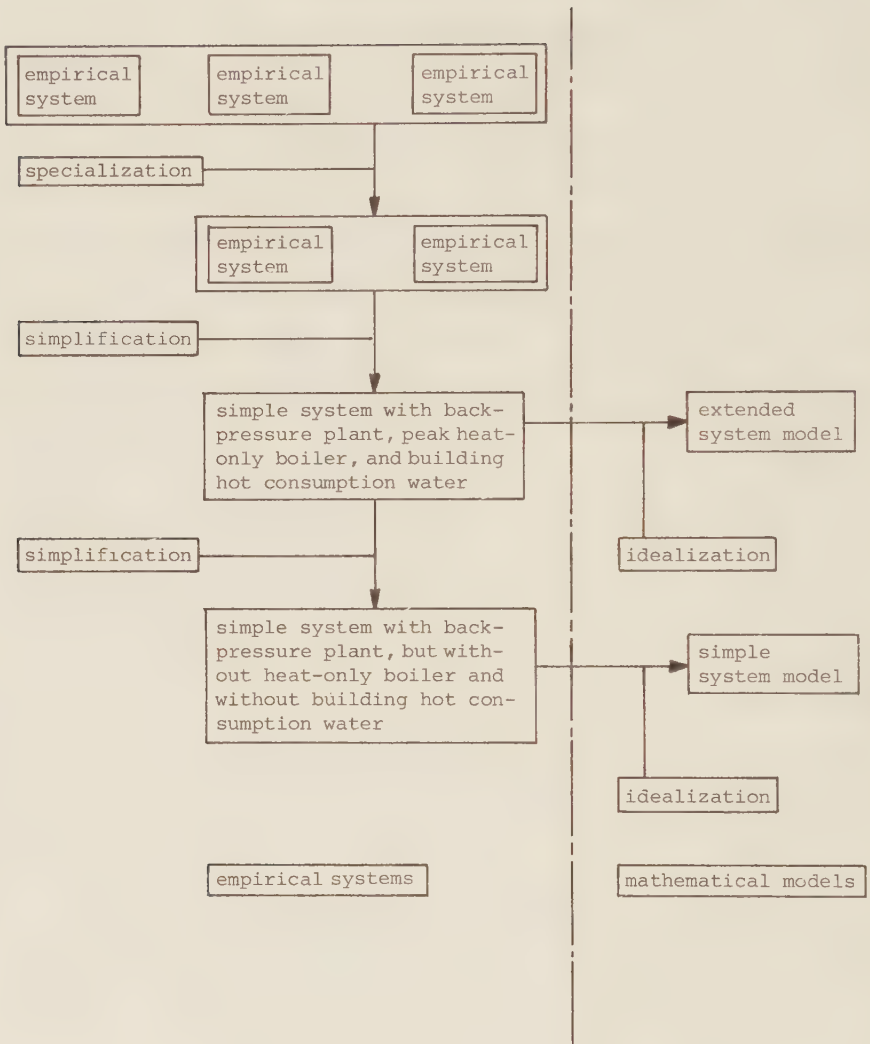


Fig. 1-4

System of concepts relating mathematical models to empirical DH systems.

step of specialization, excluding e.g. DH systems based on steam instead of hot water. From the resulting, smaller set of systems, we next carry out simplifications, such as excluding regenerative feedwater heating and steam reheating in the steam cycle. We have then arrived at the empirical counterpart of the more advanced of our two system models. By simplifying further (excluding peak heat generation in a heat-only boiler, and excluding hot water services), we get the empirical counterpart of the simplest of the two models. The two arrows pointing to the right, crossing the vertical dotted line, indicate procedures of idealisations in reducing the empirical counterparts to mathematical models.

Fig. 1-4 outlines a framework of concepts, which we shall utilize when in the next chapter we discuss our two theoretical models in relation to practical DH technology. The two theoretical models are seen to display the characteristics of being common to a whole class of empirical systems. Hence, they may be designated as generic models, a term which suggests a further feature of the models. From them it is possible to generate system variants within a special class of systems, i.e. to proceed in the opposite direction to the arrows in the figure. This feature is closely related to the aim expressed earlier, that we are seeking to explore the potential of a technology.

Generic models of the type introduced here should be distinguished from what may be designated as reproductive models, i.e. models which reproduce as closely as possible given empirical systems. Where accuracy may be seen as a central quality of this alternative type of models, generality is a more salient quality of the models we have chosen to call generic.

From the discussion above it is clear that to criticize our two models of being 'unrealistic', because they display simplifications never met in practical systems, would be inappropriate since it would presuppose that the aim of the models is to reproduce, as accurately as possible, a given empirical system. Of course this does not imply that investigations based on reproductive models as such should be irrelevant or unscientific.

By examining simple models we are able to discuss features common to a whole class of systems. In particular, we shall be able to show the implications of those technological differences distinguishing our two models, i.e. the addition of a water heating boiler and of hot water services.

In subsequent chapters we are going to explore quite extensively the simplest model first, and then we shall introduce the two types of refinement which give the more advanced, but still simple, model. Although not analysed in this work, the significance of further and, in practice, interesting complications, e.g. regenerative feed-water heating, could be examined by building still more refined models.

In fact, such a step-by-step procedure is well-known in science. In classical power cycle analysis, for instance, it is a standard method. As an example, the realistic, simple open gas turbine cycle and more complicated gas turbine cycles (with intermediate refiring, recuperation, etc) are traditionally being examined beginning with the simple cycle with constant c_p -values of air and gas, negligible isentropic losses in compressor and turbine, etc, as the first step in an analytical procedure.

1.4 Mathematical Formalism

The primary purpose of this section will be to provide the reader with a survey of the special mathematical formalism adopted in this work. As will be seen, this formalism develops as a natural consequence of the methodological approach presented in the preceding section.

When formulating mathematical equations describing the behaviour of a technological system, it will always be a matter of judgement as to how much of special formalism is appropriate. On the one hand such formalism may facilitate analytical manipulations, but on the other hand it can also cause inconvenience to the reader, not familiar with this formalism. In the present case analytical results of a fairly general nature are sought for rather extensive systems (although we shall operate with simplified models), characterized by a number of variables. Therefore, a rather circumstantial mathematical apparatus cannot be avoided in the following.

The mathematical nomenclature adopted in this work will be the result of following the three principles below:

- 1) The definition of 'reference modes' for the thermodynamical systems,
- 2) The use of dimensionless variables wherever this is appropriate,
- 3) The use of what will be designated as 'influence coefficients'.

When indexes 'o' and 'or' are attached to variables, those indexes will indicate that the numerical value of the variable is taken at a reference mode of the system. Here the term 'reference mode' may have one of two slightly different meanings, depending upon the scope of the specific investigation. The simplest case is that where index 'o' is used; 'reference mode' then means that the ambient temperature takes a reference value (0°C , as a matter of fact in our numerical examples), and that the operational mode of the DH system is fixed at a certain mass flowrate. Since our systems will have only a single degree of freedom, represented by the DH system, this is sufficient to fix the operational mode of the entire system. When instead 'or' is at-

tached to a variable, this indicates that the variable is used for an investigation where system dimensions are being varied. Then the term 'reference mode' signifies a state of the system, where both the operational mode and all dimensions (pipe lengths, diameters, etc) are taken at reference values.

Both when defining the reference mode for index 'o', and when defining it for index 'or', the actual choice of reference mode may in principle be made freely. It will be important, however, irrespective of which choice is made, that the reference mode selected is precisely defined.

The asterix * will be attached to all variables which have been made dimensionless. For instance, we shall give numerical results in the form of diagrams for the dimensionless net electric output P_{e2}^* , defined as the absolute net electric output P_{e2} divided by the heat load $P_{q2,o}$ at reference load conditions. In fact, this denominator will be used consistently when making energy flows (pumping loss P_{ec} , etc) dimensionless. The DH mass flow-rate, m , will be made dimensionless by dividing it by its value m_o at reference load. Following the practice of Anglo-Saxon literature on heat exchangers, we shall also make heat transfer coefficients (U_1^* , U_2^* , etc) dimensionless in the sense of 'Number of Thermal Units' (NTU), as defined by equations developed in section 3.3.

Symbols a, b, e, i, j, k, l_{01} , and l_{10} will be used for influence coefficients. For instance, a is defined as the relative change of the turbine output P_e per $^{\circ}\text{C}$ of change in condensing temperature at a constant steam flowrate m_s through the turbine. Thus, a is a temperature sensitivity of the plant.

One of the main advantages with the kind of nomenclature introduced here, is that it permits a substantial reduction of the number of variables necessary to describe system states, as compared to the number following from operating with variables of a more immediate physical interpretation. For instance, nowhere in system equations shall we use dimensions of turbine blades or of heat exchanger tubes. This may be seen as a consequence of our attempt to concentrate investigations to a certain level of system aggregation, as formulated in the preceding section, Method of Analysis. Turbine blade dimensions would be variables referring to a lower level of aggregation. The kind of nomenclature adopted also permits a fairly high degree of generality in

the interpretation of the calculation results. Thus, various types of heat exchanger designs (tube or plate type, e.g.) may be investigated, as far as their roles as system components are concerned, only by changing the values of one or two constants in the system equations.

When in the following chapters numerical values are selected for constants in specific calculations, it is clear that with those choices, a specialization in the validity of the numerical results will follow. It will be attempted to select constants not far from what can be regarded as normal or realistic, but the aim has been to select no more than examples of numerical values for constants. A systematic investigation of the significance of various values will be left for future investigations. However, important for the systematic of our investigations, numerical values defining reference modes, once selected, will not be changed in the course of the investigations.

A special feature of the kind of nomenclature incorporating dimensionless energy flows, is that a trivial factor of scale is eliminated. Naturally this does not imply that in practice large and small systems will display the same characteristics; various types of scaling effects will influence their characteristics. A systematic analysis of scaling effects is another point which will be left for possible future investigations.

2. MODEL FORMULATION

2.1 Presentation of Two System Models

Fig. 2-1 shows layouts of the two system models examined in this work. In the simplest model, space heat only is supplied from a back-pressure plant. The extended model incorporates in addition both hot consumption water, and a peak heat-only boiler in series with the back-pressure plant. In modern buildings hot water is nearly always needed. To design a back-pressure plant so large that it is capable of covering maximum heat demand, would require a high cost investment in peak production capacity with a poor utilization. Therefore, the extended model may be described as a more realistic, though still simplified and idealized, representation of typical empirical systems.

The hot consumption water pipeline of the extended model is supposed to incorporate a storage tank, as indicated in the figure. This feature, very common in practice, though not universally adopted, will allow us to treat the demand for heating of consumption water as a continuous load on the DH system, although hot water tapping in households is a very intermittent process.

The way the two heat generation plants of the extended model, the back-pressure plant and the heat-only boiler, have been combined represents from practice a typical solution. First, to place the two plants at a common site has practical advantages, such as common fuel storage, simplified operation, etc. However, sometimes additional peak boilers are placed at other sites between a CHP plant and the heat consumers. A typical example of this would be an oil-fired heat-only boiler serving a previously separate network, connected to a larger network incorporating a coal-fired CHP plant.

Space heating and hot consumption water represent the two most important types of heating demands in buildings in northern climatic regions, where e.g. Sweden is situated. In the USA, with its regions of hot damp summers, air conditioning (air cooling and removal of moisture from the air) is of great importance. In Central and Southern Europe cooling demands from households may be expected to increase in the years to come. Therefore, in an international context heat transportation systems, capable of providing, in addition to heating in cold seasons, cooling capacity in the summer^s, are of considerable interest for the future [2-1], [1-31]. Even in countries like Sweden, heat for purposes beyond space heating or heating of consumption

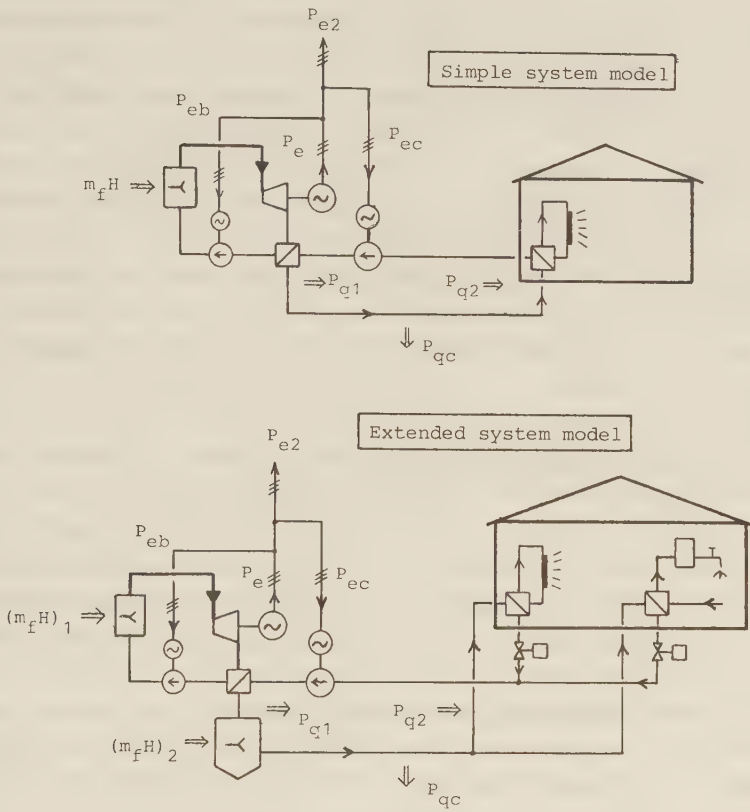


Fig. 2-1

The two simple system models examined in this work.

water is sometimes needed. Industries may have demands for various types of process heat. Drying of laundry is an example of another type of heating demand which may be present both in institutions and in private households.

In addition to the limitation of models represented by the exclusion of heating demand for purposes other than space heating and water heating, the fact that the space heating in both our models is supposed to be served by a radiator system, represents a specialization. Though it is the most common type of internal heating system for buildings, distribution of hot air has become increasingly common in later years, often as a supplement to radiator systems. Such systems, relying on forced convection instead of natural convection and radiation for the heat transfer to the interior building air, may reduce the DH supply temperature necessary at various heat load conditions. Torisson [1-23] compared the two types of space heating systems when connected to a DH system.

The type of DH technology presupposed for our two models represents a further specialization. The heat carrier is hot water, today the most common option (in Sweden the only option selected in practice), though in some countries, like the Federal Republic of Germany, steam is sometimes used instead. In the models the DH water has been separated from both the working medium of the steam cycle, and from the water of the radiator system by heat exchangers. To install a heat exchanger at the consumer's connection is standard in Sweden, but in other countries direct systems, incorporating a pressure reducing valve (and other equipment), are used instead. In this way a loss of temperature level in the heat exchanger may be avoided, so that a lower DH supply temperature may be sufficient. Advantages with a DH system incorporating heat exchangers at consumers' connections are: Freedom to choose operating pressures of the DH system independently of the durability of radiator systems, as well as improved possibilities of controlling chemistry of DH water.

In DH practice, as in our models, two pipelines, a supply and a return pipe, are generally applied. A more sophisticated system (used e.g. in West Berlin) incorporates three pipelines instead: Two supply pipes (for separate supplies of primary water for space heating and for hot consumption water) and a single return pipe (for common return). Such a system provides the possibility in the summer-time of reducing the temperature of supply water for space heating below the level needed for the water heating, as well as the possibility of

closing down this supply pipe completely, thereby reducing the heat losses in part of the year.

As a further specialization of DH technology, we shall suppose in our model calculations that the pipelines are installed underground, so that their heat losses are determined by the ground temperature, equal to the mean ambient temperature over a year, not the variable ambient temperature itself. Underground installation has been the common solution for distribution networks, while for long transmission lines installation above groundlevel with its lower investment costs has sometimes been used.

When selecting operational mode for the DH system, we shall assume that the water flowrate is freely variable (except for technological constraints, such as maximum system pressure, as discussed in section 4.2), i.e. the speed of the pump is taken to be a continuous variable. Due to the rapid implementation these years of motor thyristor control technology, this freedom in choice of flowrate now exists in many systems in practice. However, systems with discontinuous variation of the flowrate can also be found.

The back-pressure plant represents the simplest type of CHP steam cycle, and is thus a natural starting point for a general analysis. This type of plant has the advantage, due to its simplicity, of being cheaper in installation than e.g. extraction-condensation plants. The main drawback with a back-pressure cycle is that the power output is dependent upon the size of the heat load. One way of relieving this restraint is to equip the DH network with a cooling plant. The extraction-condensation plant represents a thermodynamically more advantageous way of achieving this flexibility, since here the heat removed from the cycle and not utilized for useful heating, may leave the cycle at a lower temperature level. When an extraction-condensation cycle is operated in a mode with no steam flow to the condensation turbine it acts as a back-pressure cycle, except for ventilation losses of the idle turbine in cases where this unit cannot be de-coupled from the plant alternator.

Another way of relieving the tie between heat demand and power output from a back-pressure plant is to equip the DH network with a storage tank for hot water. This is of particular interest when the CHP plant is operated parallel with an electric grid with large fluctuations in the price for electric-

ity during a 24-hour cycle. In the Federal Republic of Germany such conditions prevail [1-35]. Even in Sweden, where daily load fluctuations are covered to a great extent by hydroelectrical storage plants, hot water storage tanks at CHP plants have been found useful [1-4]. For a discussion of non-stationary operation of DH systems, the reader may be referred to Chapter 6.

As fig. 2-1 shows, the net electric output P_{e2} results from reducing the turbo-alternator output P_e by the two powers P_{eb} and P_{ec} . Further power consumptions could have been subtracted. Thus, in addition to the power P_{eb} for the boiler feedwater pump, the power station itself will consume various types of auxiliary power, usually amounting to some percent of P_e . A major post will here be the power consumption P_{ea} of the boiler fan(s) (forced draught and/or induced draught fans). Also, in the definition of the net electric output, one might include the aggregate load P_{ed} of powers for electric motors, driving pumps circulating building radiator water. The fact that such additional power consumptions have been excluded here, should be seen as a consequence of our decision, expressed already in section 1.1, to concentrate the investigations to consequences of variations in parameters of the DH system proper. When the DH operational mode is varied, the pumping power P_{ec} (following variations in DH water flowrate) will change by a much greater amount than other power consumptions of the system. In sections 3.1 and 3.5, some comments are going to be made about the two powers P_{ea} and P_{ed} , from which it will be clear that they would deserve further investigation in the future.

When system performance calculations are going to be made in Chapters 4 and 5, we shall neglect all conversion losses between thermodynamic shaft work, mechanical power, and electrical powers. Likewise, we shall neglect irreversibilities in the pressure rise of the DH circulation pump. Therefore, all results derived for DH operational modes maximizing P_{e2} will have a higher DH flowrate (and a lower supply temperature) than if the losses mentioned had been included.

In practical systems the pumping power P_{ec} may be estimated to be some 20 to 100% greater than in our theoretical models, due to the neglected losses. The pump efficiency will be in the order of 60 to 85%, varying with the operational conditions. In cases where the pump is driven by an electric motor

equipped with a thyristor control, the speed variation may cause a loss of no more than some few percent. Hydrodynamic converters are less efficient. When one or more DH pumps are located at a distance from the power plant, electric losses due to transformation and transmission may amount to 5 to 10%.

2.2 More Complicated Consumer Connections

In order to put the choice of layout for consumer connections made in the system models of fig. 2-2 into perspective, we will discuss some more complicated variants of such connections. Common to all the variants will be the existence of a heat exchanger between the DH water and the heat carrier of the internal building heating system, as presented in the last section. For a more complete treatment of variants of consumer connections the reader may be referred to [1-29].

First, we may observe that in the extended model the two heat exchangers are connected in parallel on the primary side, i.e. with respect to the flow of the DH water. This means that each of the two partial flows may be adjusted to an appropriate value by the primary control valves, for all sizes of the two types of heat demand, and for all values of the DH supply temperature, provided this temperature does not fall below certain limits, determined by secondary water temperatures and by heat exchangers.

A simple series connection of the two heat exchangers would be unsatisfactory in that such a solution would not allow for an individual adjustment of the primary flows to each heat exchanger. Therefore, if a series connection is adopted, a more elaborate solution must be chosen. Variant A of fig. 2-2 shows a layout of a solution adopted in Hungary [2-2]. The two mixing valves on the primary side make possible an individual adjustment of heat flows in the heat exchangers. Such a solution has the advantage over the parallel layout, that at high space heat loads primary water of a supply temperature in excess of that necessary for water heating, is not led directly to the heat exchanger for consumption water, but is first cooled in the other heat exchanger. At low space heat loads and a low supply temperature, a reversal of the order of the two heat exchangers seems more appropriate; a solution also discussed in the reference cited.

A mixing valve may instead be introduced on the secondary side of the connection. In variant B it has been introduced in the radiator heating system. By mixing the heated water coming from the secondary side of the heat exchanger with cooled return water from the radiators, the secondary supply temperature may be lowered, more or less, depending upon the ratio of the mass flowrates to be mixed. This type of temperature control is a standard solution for building heating systems based on individual domestic boilers. However, when

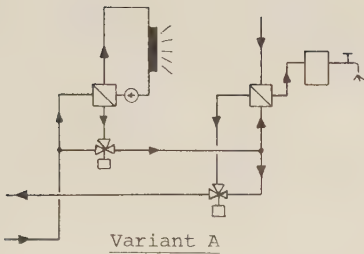
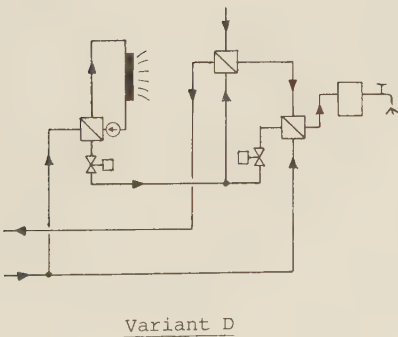
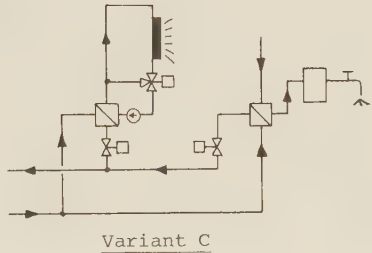
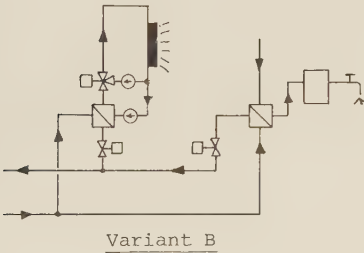
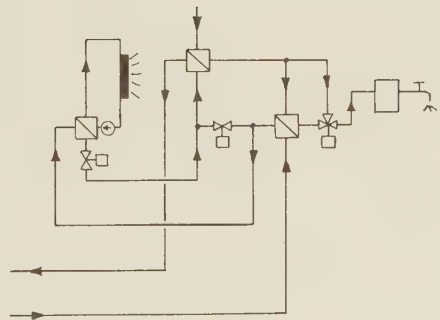


Fig. 2-2

Consumer connection variants more complicated than those adopted in simple models.



(called '2-stage' layout in Sweden)



(called '3-stage' layout in Sweden)

the radiator system is connected to a DH system, and in particular when the DH system is served by a CHP plant, this control method should be used with discretion. The temperature drop following the mixing process represents a degradation of energy, unwanted when an elevated DH supply temperature affects the generation plant adversely. An advantage with the mixing control on the secondary side is that it permits a fast adjustment of the heat supply to radiators without necessarily changing the primary flow rapidly. This may be desirable in systems where the individual heat loads of buildings change rapidly, e.g. where radiator thermostats adjust the heat supply in accordance with a time-dependable gain of solar radiation heat through windows.

In variant C a mixing valve has been inserted in the return line of the radiator system. The consequent rise in return temperature to the heat exchanger is in principle a disadvantage, since it tends to raise the return temperature of the DH system, but the mixing leads to an increased mass flowrate on the secondary side of the heat exchanger, which has the opposite effect on the primary return temperature by improving heat transfer. In some cases the latter of these two opposing effects may be predominant, so that a net gain (in terms of lower DH return temperature) is achieved.

In the simple parallel connection of heat exchangers, found in our extended system model, it is possible to cool the primary water for the water heating to a return temperature only slightly above the temperature of incoming cold water. For the primary return water from the other heat exchanger, however, a lower temperature limit is set by the return temperature of the radiator system. At high space heat loads, this temperature is normally much higher than that of the incoming cold water. Since the effective return temperature to the DH system is determined by a mixing of the two partial primary flows, this effective return temperature may be lowered by the cold water only to a limited extent in a simple parallel connection of the heat exchangers.

Variants D and E both make possible a further lowering of the effective DH return temperature. Here, the total primary mass flow passes a heat exchanger with incoming cold water. By heating the water in a second stage, where it meets primary water with supply temperature, a sufficient hot water temperature is secured for all load conditions. The two variants D and E are standard solutions in Sweden [2-3] for consumer connections in blocks of flats, where the size of the heat transferred is sufficiently large to justify the

rather high investment cost of the connection, caused by the sophistication of the arrangement. Variant D is preferred in parts of the country where unfiltered water holds a relatively large content of carbonate. Where carbonate concentrations are lower variant E, with a higher local maximum of hot water temperature in the final heat exchanger, can be used and is often preferred. This solution, with a mixing valve for the hot water, is considered to have improved control characteristics in comparison with variant D. From a thermodynamic point of view, e.g. in the sense of net electric output of a DH system served by a back-pressure turbine, a comparison of the two variants cannot be readily made; it calls for a close examination.

2.3 More Complicated Steam Plant Layouts

In this section we shall discuss some common and some less conventional complications of the simple back-pressure steam plant adopted in the two system models. The extraction-condensation cycle, which can be seen as one such complication, has been mentioned already in section 2.1. Fig. 2-3 depicts four variants which will be commented on in this section.

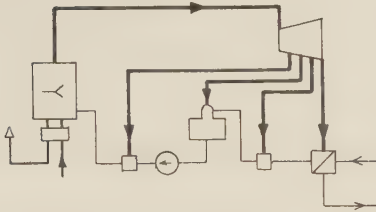
Variant A shows a back-pressure cycle with regenerative feed heating. This is a well-known method of improving the heat rate of a steam cycle. The steam bleed to the feedwater tank provides a convenient way of de-aerating the feedwater. Unless the supply temperature of the DH system is permanently high, there will be periods of operation when the steam pressure in the condenser is below the ambient pressure, so that air may leak into the cycle, making a continuous de-aeration necessary. One consequence (to be discussed in section 3.5 on the boiler) of introducing regenerative feedheating, is that variations in DH supply temperature are damped out to small variations in the incoming water temperature to the boiler.

The two-stage DH water heating, shown as variant B in the figure, is also a common cycle refinement, though not as universal as feedwater heating. For given DH supply and return temperatures, it improves the electric output of the cycle. The reason is that while in the single-stage heating all the steam flow through the turbine expands to a condensation temperature slightly above the DH supply temperature, part of the steam in the two-stage solution is condensed at a lower temperature, slightly above a level between the supply and return temperatures. Therefore, while in the single-stage case the discussion of DH temperature level influence on the steam cycle is mainly a matter of supply temperature influence, the discussion in the two-stage case must be refined to take into account the influence of the DH return temperature on the cycle.

By heating DH water in more than two stages, additional gains in electric output can be achieved. However, with increasing number of stages, the marginal gain for each additional stage diminishes rapidly.

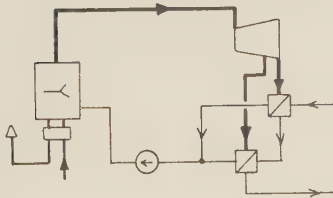
A steam driven air preheater, shown in variant C, is also well-known from steam plant practice. By adding this component to the air preheater driven by combustion gas, the risk of low temperature corrosion (cf. section 3.5) is

Fig. 2-3 Generic back-pressure steam plant layouts; variants more complicated than that adopted in the two system models analysed in the thesis.



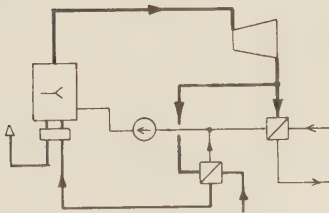
Variant A

Regenerative feedwater heating



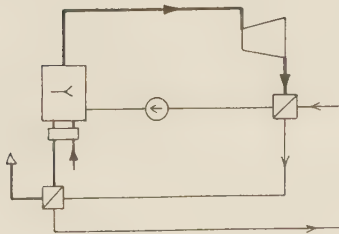
Variant B

Two-stage heating of district heating water



Variant C

Steam driven air preheater



Variant D

Addition of heat to district heating water from combustion gases

reduced. At low ambient temperatures the steam flow for preheating is increased, so that variations in the ambient temperature are only partly transferred to variations in the temperature of the air incoming to the preheater exposed to corrosive combustion gas. In the figure the steam for the preheater is taken from the turbine exhaust. This is thermodynamically advantageous compared to utilizing extraction steam. In the case of a condensing plant, however, extraction steam must be used, since the exhaust steam in that case has too low a temperature. It is interesting to note that in the case of steam extraction from a back-pressure plant, as in the figure, the electric output from the turbine increases at such load conditions where the steam flow through the turbine is less than maximum. This is one additional advantage of a steam driven air preheater for a back-pressure plant, as compared to a condensing plant.

In the last layout of fig. 2-3, variant D, a heat exchanger is added which transfers heat from the combustion gases directly to the DH water. This less conventional solution may be attractive in the case of combustion gases with a low corrosiveness, making possible a utilization of water vapour condensation heat. One instance of this would be when natural gas is used as fuel. Also, combinations with scrubbers for removal of sulfur dioxide are possible. Since the specific heat of combustion gases increases significantly below the water dew point, only a minor part of the water condensation heat could be recovered by the air to the boiler (the air has a low value of the product of specific heat and mass flowrate). The DH water, with its higher mass flowrate, is more suited to recovery of condensation heat.

The four variants of fig. 2-3 are basic options, from which many combinations may be generated. For instance, it is common practice to combine variants A and B, so that steam flows for DH water heating are derived by branching off from extraction flows for regenerative preheaters. In this way the number of extractions from the turbine can be limited.

In all the variants of the figure, only a single turbine has been shown. As with condensing plants, improved efficiency for electricity can be achieved by introducing reheating of steam. This is of particular interest in the case of larger plants, where complications of the plant are more readily motivated in an economic optimization.

2.4 Single Heat Load Versus Divided and Distributed Load

In both the simple system models examined in this work, the heat load is represented by a single building. This is very much simplified in comparison with DH systems found in practice. Here, the heat load is an aggregate of many (sometimes more than 1000) smaller loads, and the pipeline system is a branched network instead of the single pair of pipes in the models. However, as we shall find in this section, the models may be expected to reproduce with reasonable accuracy some types of parameter variations in branched networks, while other variations performed in real systems are beyond the scope of our simple models.

In the discussion below we shall at first point out some important differences in the characteristics of the simple models and of branched systems. This will enable us to examine to which extent our models can be expected to reproduce branched system performance, first in the case of pure load performance (section 4.1 and Chapter 5), and then variations in system dimensions and in topology (section 4.3).

Some properties of branched systems

For the comparison of simple models with branched systems, the following differences will be of importance:

- A) In our models, the DH pumping power P_{ec} is the sum of losses developed in pipes and in heat exchangers. In branched systems, a substantial part of the pumping power will in addition be caused by pressure losses in valves.
- B) In our models we shall be careful to take into account the variations in supply and return temperatures from heat generation to its delivery. In branched systems, different consumers are in general located at various distances from heat generation, so that the temperature variations along the pipes will not be the same for all consumers.
- c) So long as we assume that in the simple models DH pipeline cross-sections are the same at all points between heat generation and delivery, heat and pumping losses will be distributed uniformly along the pipelines. In branched systems, in contrast, the two

types DH thermodynamic losses are normally distributed in a highly non-uniform way.

- D) In branched systems, individual heat consumers may have heat exchangers with various types of difference in performance characteristics.

Below we shall explain and analyse these various differences in more detail.

Fig. 2-4 is a schematic of the main structure of a typical DH system, with a single heat generation plant in the middle. At many points of the network consumers are connected at different distances from the plant. It will be important to distinguish between geographical distance and hydraulical distance, as indicated by the figure. As is seen, the network employs several loops, i.e. the flow may arrive at a consumer along different routes. Here, the hydraulical distance must be defined specifically in each particular case. Loops are not mandatory, but are frequently present in large DH systems, where the flow path redundancy is believed to increase reliability of heat supply. Another reason for introducing loops may be that bottlenecks in the flow area of pipes, created during the expansion of the network over the years, may be relieved.

In a branched network, pipes of many sizes are present. On the top of fig. 2-5 is shown in principle, how the pipe diameters decrease from the generation plant to a distant consumer. As indicated in the diagram below, the water pressure of the supply pipe at the same time gradually goes down. In the diagram an approximately constant pressure gradient has been assumed. In DH practice it seems to be common to design for pressure gradients approximately independent of pipe diameter. However, due to the fact that consumers are normally connected to a DH system by a gradual procedure over the years, the pressure gradient in a given pipe may change significantly with time, and very different gradients may be present in different parts of a system at a given stage of expansion.

The pressure gradient along a pipeline may be written as:

$$\frac{dp}{dx} = \frac{\lambda}{D} \frac{1}{2} \rho v^2 \quad (2-1)$$

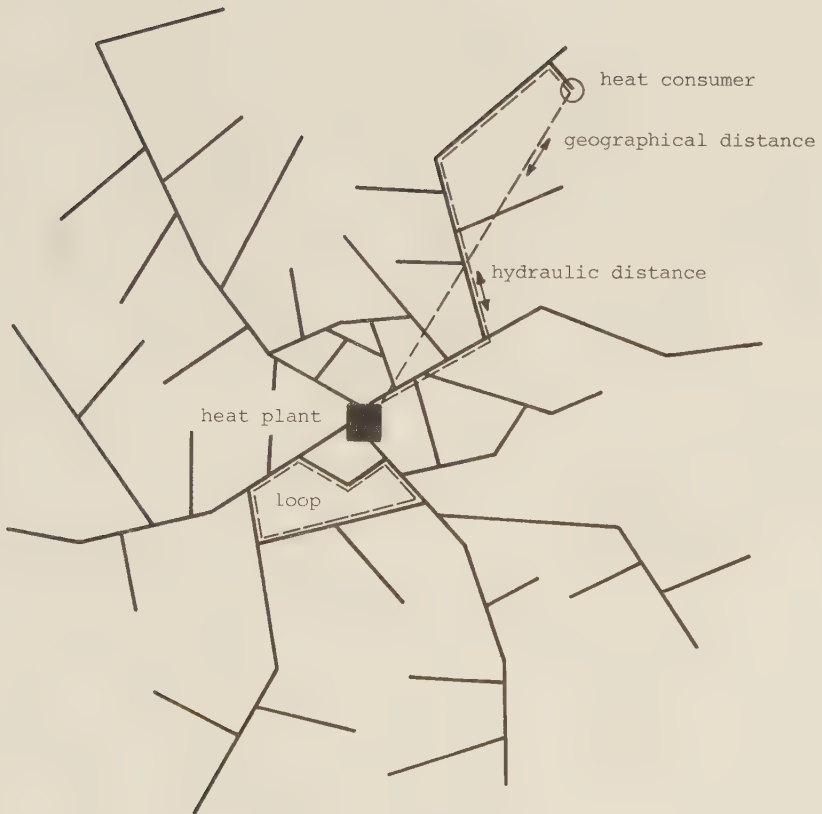


Fig. 2-4 Branched DH system (main pipes only)
with heat generation plant at centre.

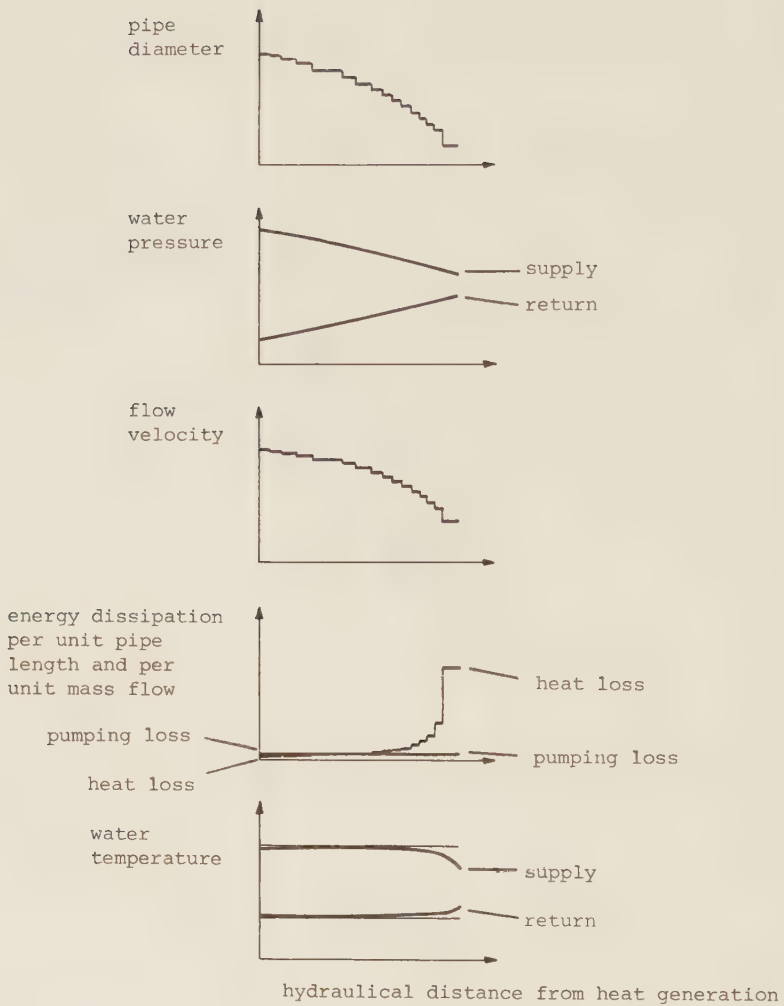
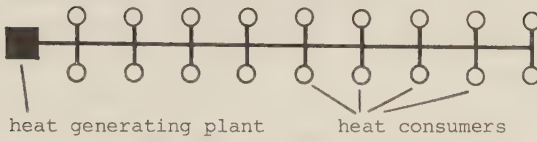


Fig. 2-5 Variation of pipe diameter and of thermodynamic parameters in DH network along a flow pathway from heat generation to a distant consumer.

linear network configuration:



network configuration with equal distances between heat generation and all consumers:

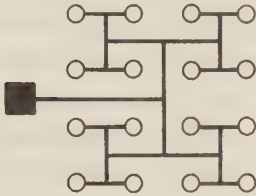


Fig. 2-6

Two types of theoretical, regular DH network structure.

pressure holding vessel

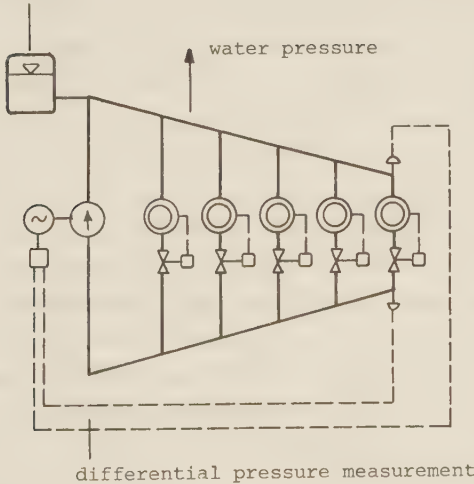


Fig. 2-7

Pressure control in DH system.

Thus, a constant pressure gradient means that the velocity v must go down with reduced diameter, as shown in the third diagram of fig. 2-5. The diagram below shows the variation of heat and pressure losses from heat generation to the consumer, expressed as energy dissipation per unit pipe length and per mass flow unit. For the pressure loss, the dissipation rate is approximately constant, while the heat loss increases rapidly with reduced diameter, due to a deteriorating volume/surface ratio for the pipes.

From the variation of energy dissipations the variation of supply and return temperatures can be estimated, as shown in the next diagram. It is seen that over a long distance from the generation plant the variation is small, while at the end of the system the supply temperature drops rapidly. Close to the plant the supply temperature is seen to increase slightly in the flow direction, due to the fact that here the pressure energy dissipation has been assumed to be larger than the heat dissipation, as indicated by the diagram above. In practice this is not always so; in smaller DH systems there will nearly always be a net temperature drop in the supply pipes in all parts of the system.

In the case of a DH system with loops, as in fig. 2-4, the abscissa in fig. 2-5 may follow different routes from heat generation to a given consumer, as pointed out before. In pipes belonging to loops the flow may be small (the particular pipe acts primarily as a flow reserve), so that the pressure gradient will also be small. For such pipes the heat losses may dominate over energy dissipation due to pressure losses.

In a DH network, consumers at various distances from the generation plant are connected to points with very different pressure differentials between supply and return lines. In the simple models the single heat consumer is in contrast exposed to a pressure differential amounting to no more than a fraction of the value at the generation plant. For consumers at a short distance from the generation plant of a network, it would be possible to design heat exchangers for large primary side pressure drops, thereby 'utilizing' the pressure differential instead of dissipating it in valves. However, heat exchangers are very much standardized, so that this is hardly done in practice.

The fraction of the total pumping power dissipated in primary control valves varies with network structure. Fig. 2-6 shows two theoretical, extreme cases.

In the linear arrangement roughly half of the pumping power is dissipated in control valves. In the network below all consumers are located at equal flow distances from the generation plant. Here, the fraction may be very small. For practical networks the fraction can be expected to lie somewhere between 0 and 0.5.

So far we have assumed that the pumping power is added to the system at a single point close to the generation plant. By dividing the pump into several units, and placing them at different locations of the network, it is possible to reduce the maximum pressure differential between supply and return lines, and thereby the fraction of the total pumping power dissipated in primary control valves. In practice such additional pumps are often found at locations where water is flowing into previously separate networks, later connected to a larger network. Unfortunately the principle of dividing up the pump cannot in practice be carried very far, since the number of pumps needed for this would increase rapidly, as in a network the number of branches increases with the distance from the generation plant.

For the purpose of compensating for variations in primary differential pressure with geographical distance, valves with fixed settings would suffice. However, in practice a variable setting is necessary to make possible corrections in allocations of primary flows to different consumers. The control range of the valves should not be too restricted, so that pressure drops over the control valves cannot be small in comparison with pressure drops over corresponding heat exchangers. This means that even for the most distant consumer a certain minimum pressure differential between supply and return pipes must be assured.

Fig. 2-7 shows a common way of controlling the pressure in the pipelines of a DH system. One point of the system, in the figure the point of maximum system pressure, is selected for a fixed pressure control. The speed of the circulation pump is governed from a measurement of differential pressure at a suitable location in the system. In the figure the location of the consumer at greatest distance from the pump has been selected for this. Hereby the overall mass flowrate is determined at a given setting of primary control valves and for a given supply water temperature. The settings of the primary control valves for the various consumers are given by signals indicating the individual needs for primary flows, varying with the supply temperature.

In networks where it is certain that the minimum differential pressure is found at the point of measurement, sufficient differential pressures at all locations can be assured simply by keeping the measured differential at a satisfactory level. In some networks, however, two or more points at equivalent flow distances from the circulation pump may be candidates for the role of minimum differential pressure. The location of this minimum may even shift from one load condition to another. In such cases a solution may be to select one point for measurement as in fig. 2-7, but set the measured differential pressure with a safety margin, so that at no point in the system does the differential pressure fall below what is acceptable. This safety margin results in a greater pumping power in primary control valves. A more refined solution would be to measure differential pressures at all points of the possible minimum value and, according to which location has the actual minimum value at each load condition, that value is selected for pump speed control by a control unit, capable of performing this logical operation.

Load performance of simple models and of branched systems

In the case of a theoretical network like that in fig. 2-6, where all consumers are placed at equal flow distances from heat generation, it is clear that a model with a single heat load may be an adequate representation, provided all consumer connections respond in the same way to load change. This condition may be satisfied in several ways. The simplest case is that of identical individual heat loads. Another case is that where individual heat loads are of different sizes, but where heat exchanger areas and primary flows are scaled to the load sizes, so that at all load conditions the primary flows to different consumers are cooled to the same return temperature.

Variations in sizes of building wall insulations among different consumers in a network are unessential to the question of simple model validity, so long as only space heating is present and the network is of the idealized type shown at the bottom of fig. 2-6. In the more realistic case of hot water production along with space heating, differences in the sizes of insulation will normally result in different ratios between the two types of heat load among consumers, so that variations in individual return temperatures are created. Since this will affect heat loss components associated with return lines, the simple model will not in a strict sense be a correct representation of such cases. However, if the ratio between the two heat load types of the model

equals the corresponding ratio for aggregate loads of the network, the model will still be a good approximation.

In practical systems return temperatures from different consumer connections may differ for many reasons. Initially equal connections may alter over time because of unequal fouling of heat exchangers. Small heat exchangers are often designed with higher return temperatures than larger units, taking advantage of size effects on specific investment costs. Thus a simple model representation of a network will never be more than an approximation, based on 'mean' consumer connections, with a number of network effects giving deviation between reality and model.

In the case of the idealized network in fig. 2-6 all consumers may be assumed to receive supply water of the same temperature. In a realistic network, consumers are situated at different hydraulic distances from heat generation, so that flows to various consumers are subjected to unequal temperature drops. As a consequence, the size of a primary flow necessary to satisfy a given heat requirement to a given heat exchanger increases with distance from heat generation. Also, the temperature drops for various partial primary flows will change with load conditions.

In the simple system model with space heating only, the DH pumping power is assumed to be proportional to the third power of the mass flowrate, since all flow resistances are unchanged with load changes, and turbulent flow conditions may with reasonable accuracy be assumed to prevail. In the load performance calculations for the simple model variant with hot consumption water production, we are going to assume that flow resistances in primary valves (determining the distribution of flows to the heat exchangers for space heating and for hot water production) have negligible flow resistance. Therefore the DH pumping power may also for this variant be assumed to vary with the third power of the total DH mass flowrate. If the simple models are to be reasonable representations of realistic networks, the relation between pumping power and flowrate for the networks cannot be allowed to deviate much from the relation valid for the models.

To examine this, let us first look at a hypothetical DH system with or without loops, where all flow resistances in system components are constant, i.e. positions of valves are kept fixed. The circulation pump may be divided into several units, but if they are, the pumps must be controlled in a way to as-

sure that pressure rises in the pumps change in the same proportion when the operational conditions for the system are altered. For such a network, all mass flows in different branches will change in proportion to the overall mass flowrate, and all pressure drops will be proportional to the square of mass flows, equivalent to a change of pumping powers with the cube of mass flows. It is seen that a system with the type of control shown in fig. 2-7, capable of adjusting individual massflows in accordance with exact changing demands for heat supplies, will be an approximation to the idealized network just described, if the differential pressure governing the circulation pump is changed with the square of the overall mass flowrate. Then only minor adjustments of valves will be necessary at changing load conditions.

In practice the control will normally be designed for a fixed differential pressure at the point of measurement, but so long as this constant differential pressure is only a fraction of the maximum differential pressure of the system, pressure drops in various parts of the system will still change approximately in proportion to the square of the overall mass flowrate. Sometimes circulation pump controls are governed by measurements of differential pressures somewhere between heat generation and the consumer at greatest distance [2-4], keeping the value constant. In such cases the pressure drops will change less with overall mass flowrate, and the simple models will not be good approximations to the networks with changing load conditions.

The conclusion of these qualitative considerations is that our simple models need not be poor representations of realistic, branched networks. Individual variations in heat demands for different consumers, along with other phenomena, will give deviations between reality and the simplified models, but with the right type of overall network control, the simple models may be considered as reasonable first approximations of reality for load variations.

Variation of design parameters in simple models and in branched systems

First we may establish that changes in diameter and in length of selected branches in a network can not readily be reproduced by the simple models. Due to the uneven distribution of heat and pumping losses, as shown in fig. 2-5, such changes in system dimensions will affect system characteristics very differently, depending on how far from heat generation the branch in question is situated. For instance, an increase in the diameter of a large diameter pipe close to heat generation may lower the total pumping loss significantly, with

only a slight increase in heat loss. If the simple models, with constant diameter from heat generation to heat delivery, are to reproduce such network variations, elaborate definitions of equivalent diameter, length and insulations would be called for.

Direct counterparts in branched networks to changes of pipe dimensions in the simple models are changes which affect all branches in an equal way. For instance, two variants of system models with different pipe lengths may be seen as representatives of two branched systems serving built-up areas with different area densities for heat consumption, so that all pipe lengths in the one network are longer than those of the other network by a constant factor. For a further discussion on such changes in system dimensions the reader may be referred to section 4.3.

Fig. 2-8 illustrates a special type of effect of changes in pipe diameter of branched systems. In the schematic to the left, the pressure gradient has been assumed to be the same throughout the system. Since end points of side branches are placed at varying distances from the generation plant, a consequence of this pressure distribution is that for the nearest branches more pumping power is throttled away in valves than for the branches far from heat generation. If pipe diameters in branches are instead selected to give a varying pressure gradient, as in the schematic to the right, the fraction of total pumping power lost in valves will go down, and the fraction lost in pipes will at the same time increase. The size of the total pumping power may hereby remain constant. In this way reduced investment costs for pipes, and reduced heat losses, may be achieved without affecting the size of the pumping power. This kind of effect is not found with the simple models, where no pumping power is lost in control valves.

It is clear that in general changes in network topology cannot be reproduced in the simple models. This does not mean, however, that results found for the simple models are irrelevant for considerations of this type of change in real networks. Consider, for instance, the case of a given network designed for heat supply from a heat-only boiler. At the introduction of a back-pressure plant it may be considered to lower the supply temperature, since this will result in a higher electric output, as calculated from a simple model. Such a lowering of temperature may be done without resulting in an unacceptable increase in pumping power, if the flow resistance of the network is made

Case of constant
pressure gradient:



Case of equal pressure diffe-
rentials in branch end points:

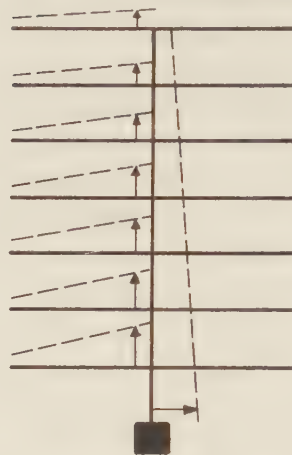


Fig. 2-8 Alternative practices for pressure gradients in side branches.

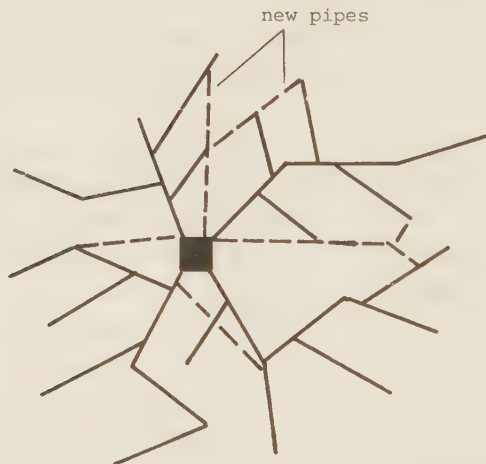


Fig. 2-9 Modification of network structure by adding pipes to create a system with loops.

smaller. In principle this could be done by increasing pipe diameters, but for pipes installed already this may not be attractive. A more practical alternative may be to install additional pipes, as indicated in fig. 2-9. It is seen that, as a result, a network with more loops is created.

3. CHARACTERISTICS OF SYSTEM ELEMENTS

3.1 Building Heating Requirement and Internal Heating System

Building heating requirement

An accurate thermal representation of the complicated subsystem of one building with several rooms or of a group of buildings, would call for a detailed study involving variations in construction principles, internal heating system, etc. Here, we shall be satisfied with a very simple representation of the subsystem.

First, we shall exclude cases with an ambient temperature t_a above the room temperature t_h , i.e. we are only concerned with heating of buildings, not with cooling or air conditioning, as discussed in section 2.1.

To be able to represent variations in heat requirement in a simple formula we shall make the following assumptions:

- The heat requirements of the building are assumed to vary in proportion to the difference, $t_h - t_a$, between in- and outdoor temperatures,
- The room temperature, t_h , is assumed to be the same in all parts of the building, and t_h is assumed to be constant, irrespective of variations in ambient temperature and operational strategy for the DH system,
- Possible internal heat gains from lighting, cooking appliances, etc, are neglected.

With these assumptions the dimensionless heat load to the DH system,

$P_{q2}^* = P_{q2}/P_{q2,o}$, may be written as:

$$P_{q2}^* = 1 + i(t_h - t_{a,o}) \quad (3-1)$$

where the constant i is:

$$i = 1/(t_h - t_{a,o}) \quad (3-2)$$

A linear type of relationship between heat requirement and ambient temperature is expected from the fact that heat conductivities of both building materials and of insulation materials must be relatively constant at the moderate temperature variations, to which they are normally subjected. The degree-day concept, universally accepted as a basis for estimations of yearly heat requirements of buildings, also presumes a linear kind of relationship. Measurements of variations of heat loads for DH systems [3-1], [3-2], indicate that as a statistical mean the linearity holds, not only for single buildings, but also for groups of buildings, connected to a DH system.

When daily heat loads for a DH system are plotted against the daily mean of the ambient temperature, a scattering of several degrees centigrade between single days of equal heat loads will appear. One reason for this is that secondary factors, such as wind speed, solar radiation on buildings, varying from day to day, all have some influence on the heat demand. Another reason is that the heat storage capacity of building walls may be large enough for the heat load of one day to influence the heat requirement of the subsequent day significantly, the time constant of buildings is typically in the order of one or a few days. The heat storage effect is also reflected in the fact that when designing radiator systems, the maximum heat requirement of a building is calculated according to a lower design ambient temperature for a light building than for a heavy one [3-3].

In calculations of degree-day units, according to the various national standards, ambient temperatures are subtracted from, not a typical room temperature, but a temperature some degrees below, e.g. 16°C . This is a practical way of taking into account heat gains from housing appliances, hot water in buildings, etc. An accurate determination of the numerical relationship between heat requirement of a DH system and the ambient temperature, would call for a more individual assessment of such heat gains. The general tendency in building construction to increase insulation, along with the growing number of household electric appliances, both act in the direction of increasing the relative importance of heat gains. Increased demands for exchange rates for indoor air will act in the opposite direction.

In building heating practice the level of the indoor temperature t_h , considered as a constant in our model, may vary for different reasons. Until recently radiator thermostats were not very frequent, and with hand operated valves instead it is clear that the adjustments of room heating to changes in actual heat load will be rather inaccurate. On the other hand, improved possibilities of controlling indoor temperatures make possible deliberate variations in indoor temperatures, such as a lowering of the temperature level at night. The impact of such variations will be outside the scope of our treatment, but a short discussion of non-stationary effects is given in Chapter 6.

Thus, the linear relationship between building heating requirement and ambient temperature, assumed for our model calculations, may be characterized as reasonable first approximation. As we shall discuss in section 4.1, controls for DH systems will have to take into account the statistical variations from the mean linear relationship. In more accurate model calculations, one might also consider adopting, as a substitute for the linear relationship, a piecewise linear relationship, as suggested by Munser [1-1].

Building heating system

As outlined in section 2.1, the transport of heat from the DH consumer connections to the air in the rooms of the buildings is assumed to take place via a conventional central heating system with radiators, see fig. 3-1.

For radiators, heat transport from the metal surface to the air outside the thermal boundary layer is composed of natural convection and radiation. For natural convection, the heat transfer coefficient according to conventional heat transfer theory is included in the Nusselt number, obeying a relationship of the following type:

$$Nu = \text{constant } Gr^p Pr^q \quad (3-3)$$

where the Grashof number is defined as:

$$Gr = \frac{g\beta\Delta L^3}{\nu^2} \quad (3-4)$$

Here, g is the gravitational acceleration, β is the thermal expansion coefficient for air, θ is the driving temperature difference between metal surface and air, L is a characteristic length and ν is the kinematic viscosity of the air. The Grashof number in the formula above replaces the Reynolds number in the case of forced convection, instead of natural convection.

The exponent p for the Grashof number is typically around 0.25, which means that the heat transfer coefficient improves somewhat with increasing temperature difference. Also for the other type of heat transfer mechanism active with radiators, thermal radiation, there is an improvement of heat transfer with increasing temperature difference.

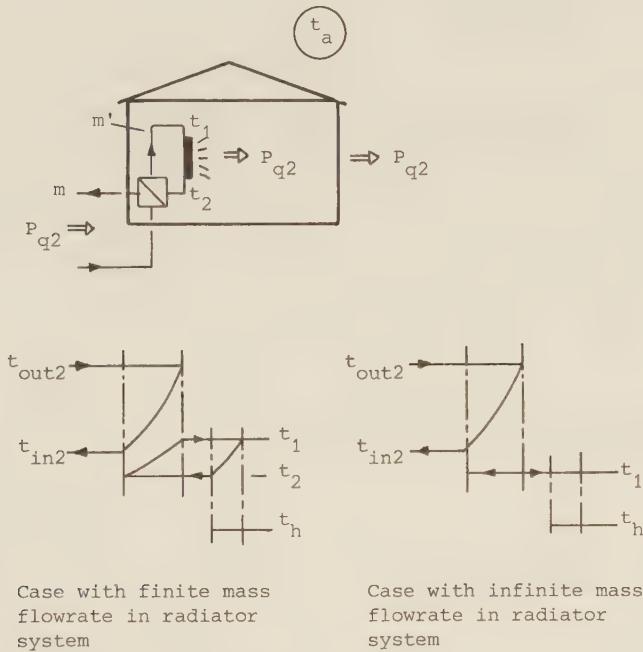


Fig. 3-1

Consumer's heat exchanger and internal building heating system of simple system model.

This overall heat transfer characteristic of radiators may be contrasted with the case of forced convection over DH-air heat exchangers, where the heat transfer coefficient depends primarily on the air velocity over the heat transfer surface, whereas the temperature influence is only minor.

The calculation of the heat transfer for a radiator is simplified by the fact that the heat resistance on the air side dominates in comparison with the resistance on the water side and in the wall material. The formula for the Nusselt number cannot, however, be applied directly, as the driving temperature difference in general varies over the surface, following the variable water temperature at a finite water flow. According to the standard literature [1-27], the overall heat transfer (including radiation) is with reasonable accuracy given by the formula:

$$P_{q2} = U_3 A_3 \theta_3 \quad (3-5)$$

where U_3 is the heat transfer coefficient, A_3 is the surface area, and θ_3 is the logarithmic mean temperature difference, in turn defined as (conf. fig. 3-1):

$$\theta_3 = \frac{t_1 - t_2}{\ln \frac{t_1 - t_h}{t_2 - t_h}} \quad (3-6)$$

The variation of P_{q2} is given by:

$$P_{q2}^* = (\theta_3 / \theta_{3,o})^n \quad (3-7)$$

(typical value: $n = 1.3$)

In the model calculations we shall introduce the simplification:

$$t_1 = t_2 \quad (3-8)$$

i.e. constant temperature of the radiator water, corresponding to an infinite mass flow in the radiator system. Also, in spite of this great flow, we shall assume that the power for circulating the radiator water is negligible.

This simplification is reasonable to make as a first approximation, since the temperature drop of the radiator water is usually moderate in comparison with the mean temperature difference between radiator water and room air, and also in comparison with normal temperature drops of primary DH water in a substation. In section 3.3 we shall examine effects of varying temperature differences between radiator supply and return temperature.

With the assumption made above the (constant) radiator temperature can be given as follows:

$$t_1 = t_h + (t_{1,o} - t_h) P_{q2}^*{}^{1/n} \quad (3-9)$$

3.2 Heat and Pumping Losses in the DH System

Application of the First Law of Thermodynamics to the DH system gives:

$$P_{q1}^* = P_{q2}^* + P_{qc}^* - P_{ec}^* \quad (3-10)$$

Normally the heat loss P_{qc}^* is much larger than the pumping power P_{ec}^* , so that the heat input P_{q1}^* to the system exceeds the heat requirement P_{q2}^* of the building(s). In special cases, such as when the DH system includes long transmission lines, P_{ec}^* may be larger than P_{qc}^* .

Supposing that the geometrical properties of all parts of the DH system (including possible valves) are not changed with operating conditions, and assuming fully developed turbulent flow, the pumping power at static operation conditions varies according to the formula:

$$P_{ec}^* = P_{ec,o}^* m^3 \quad (3-11)$$

The total pumping power P_{ec}^* may be broken down into four components, given by coefficients x , y , z , and w , for the simplest system model;

$$x + y + z + w = 1 \quad (3-12)$$

where x is the part of P_{ec}^* dissipated in the supply line, y the part in the return line, and z and w the parts dissipated in the heat consumer and power station heat exchangers respectively. x , y , z , and w may for our simple model calculations be taken to be constants at variations in operating conditions, so that:

$$zP_{ec}^* = zP_{ec,o}^* m^3 \quad (3-13)$$

and similar formulae for x , y , and w .

The variation of the heat loss P_{qc}^* with operation conditions is somewhat more complicated. A number of simplified treatments in literature on heat losses from DH systems, give the loss as a function of the supply temperature only, for different types of pipeline cross-section geometry. This may

be sufficient in cases where the heat input is supplied from a heat boiler only; here the boiler efficiency is practically unaffected by supply and return temperatures. With a back-pressure turbine, however, those temperatures are important for the plant performance, and since changes in supply and return temperatures along the pipes are determined by, among other factors, how the heat loss is divided between supply and return lines, such a distinction will be necessary for a stringent analysis.

The discussion below will show that it is suitable to represent the heat flow situation around a pair of underground DH pipes by three separate superimposed heat flows, namely:

- a flow $P_{q_{IO}}^*$ from the supply line to the environment,
- a flow $P_{q_{IIO}}^*$ from the return line to the environment, and
- an internal flow $P_{q_{III}}^*$ from the supply to the return line,

a kind of representation presented already in [1-24]. Although it is both convenient and, as we shall see, not confined to some special type of pipe configuration, it is, somewhat surprisingly, not mentioned in the majority of the treatments on heat losses given in the literature. A recent work, Zeitler [3-5], touches upon it, but although careful to distinguish between influences on heat flows from supply and return lines, he is concerned primarily with arriving at accurate formulas for the total heat loss rather than for temperature variations in supply and return lines.

Before considering total heat losses along pipes, it is convenient to examine heat losses per unit length of pipe. Fig. 3-2 shows in principle a circumferential distribution of heat loss, q'' , per surface square unit. By performing contour integrations:

$$q'_1 = \oint q''_1 ds \quad (3-14)$$

$$q'_2 = \oint q''_2 ds \quad (3-15)$$

for supply and return lines, respectively, heat losses per unit length of pipe are derived.

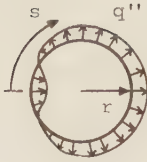


Fig. 3-2

Variation of heat flux q'' per surface unit as a function of the peripheral coordinate s .

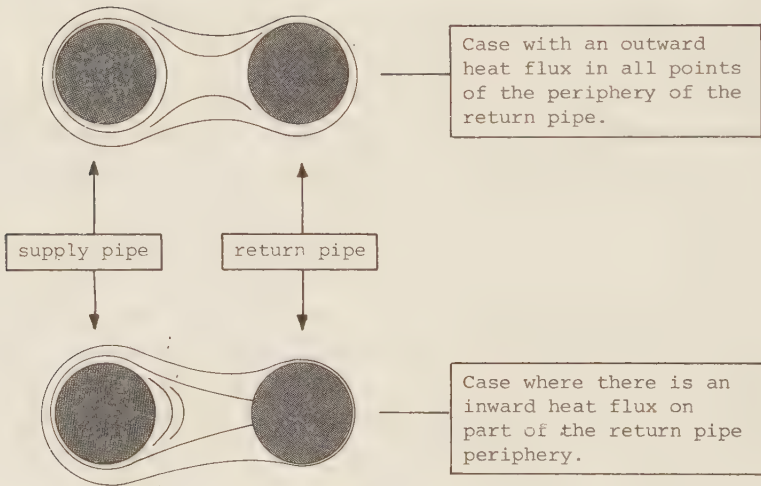


Fig. 3-3

Isothermals round a pair of DH pipes underground.

The heat fluxes q_1'' and q_2'' per unit surface area may be read from plots of isothermal lines [3-6] around the pipes, as shown in fig. 3-3. In case A all isothermals near the return pipe are closed curves, containing the pipe. Between the two pipes there is a region where lines connecting the pipes have a minimum, while on the lines in the perpendicular direction there is a maximum. At the centre we have a saddle point. In contrast to this case B shows a situation with no saddle point and with two isothermal lines ending at right angles on the surface of the return pipe. Thus, in case A the heat flux q_2'' from the return line is positive at all points of the periphery, whereas in case B q_2'' is positive only to the right of the two isothermal end points. To the left of these there is a local influx of heat to the return line. In spite of this, the integrated heat flux, q_2' of the return line may very well be positive in case B.

The heat fluxes q_1' and q_2' per unit length of pipe produce changes in the temperatures of the two flows in the pipes, in combination with the temperature changes due to internal heat generation from pressure losses of the incompressible fluid [1-24], [3-7]. For the supply line we have:

$$\frac{dt_{out}}{ds} = - \frac{q_1'}{mc} - \frac{1}{cp} \frac{dp}{ds} \quad (3-16)$$

giving the temperature gradient in the flow direction. Here, c is the specific heat capacity of the fluid. Since q_1' is positive, and dp/ds is negative, the gradient may be either positive or negative, depending upon the magnitudes of the two terms on the right-hand side of the equation. Normally the heat loss term will dominate, so that there is a net temperature drop along the supply line in the flow direction. Observing, however, that the pipe perimeter is proportional to the diameter, and that the mass flow-rate is proportional to the square of the diameter at constant flow velocity, we find that there will be a tendency for the heat loss term to decrease with increasing relative size of heat transported by the pipeline system. It can therefore not be excluded that at sufficiently large heat transports there will be a net increase of supply water temperature in the flow direction [3-8]. Because of the risk of water boiling, in such cases there will be an upper limit to the permissible length of the system. This kind of risk may be even more accentuated in the return line, where the

heat flux per unit length, q'_2 , may be negative, i.e. there may be a net influx of heat to the fluid.

Our investigation here will specialize in static heat flow conditions. Although there is both a 24-hour variation and a yearly variation of earth surface temperature, both of substantial magnitude, those variations decrease rapidly with depth below the surface, so that at typical pipeline depths of about 1 m there is only a slight temperature variation. The heat loss components will therefore be relatively unaffected by changes in earth surface temperature. Time-dependent variations in DH water temperatures may have greater effects upon the temperature field round the pipes. However, in most cases the pipelines are so well insulated that the main heat capacity (that of the surrounding soil) is found where the temperature variations are small. Homonnay [1-28], [3-9] has discussed non-stationary heat conduction phenomena referring to DH pipelines.

With t_{out} being the temperature of the supply line flow medium, t_{in} the temperature of the return line, and t_s being the (time-average) earth surface temperature, the two heat losses per unit length of pipe may be written as:

$$q'_1 = c_{11}(t_{out} - t_s) + c_{21}(t_{in} - t_s) \quad (3-17)$$

$$q'_2 = c_{22}(t_{in} - t_s) + c_{12}(t_{out} - t_s) \quad (3-18)$$

where the four coefficients, c_{11} , c_{21} , c_{22} , and c_{12} , characterize the heat conduction properties of the configuration. The coefficients will be constants, so long as at each point of the thermal field the heat conductivity is constant. Normally heat flow resistances of the insulation material will dominate in comparison with the resistance of the soil. Variations in soil moisture content may lead to great variations in the conductivity of the soil, while for the insulation material the variation is normally small or moderate. In the case of mineral wool the conductivity will normally vary by 10-30% when the water temperature changes. However, in the case of wetting of the insulation material at water leakages the heat conductivity may be subject to great fluctuation.

In case $c_{21} = c_{12}$, the two equations may be rewritten as:

$$q'_1 = (t_{\text{out}} - t_s)(c_{11} - c_{12}) + (t_{\text{out}} - t_{\text{in}})c_{12} \quad (3-19)$$

$$q'_2 = (t_{\text{in}} - t_s)(c_{22} - c_{12}) - (t_{\text{out}} - t_{\text{in}})c_{12} \quad (3-20)$$

Putting:

$$R'_{10}{}^{-1} = c_{11} - c_{12} \quad (3-21)$$

$$R'_{20}{}^{-1} = c_{22} - c_{12} \quad (3-22)$$

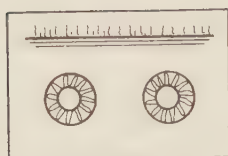
$$R'_{12}{}^{-1} = c_{12} \quad (3-23)$$

it can be seen that eqs. (3-19) and (3-20) describe the heat flow of the network model at the bottom of fig. 3-4. The three resistance values R'_{10} , R'_{20} , and R'_{12} may be calculated by various methods for different types of pipeline configurations, of which the figure shows some examples.

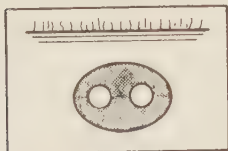
In the example of two separately insulated pipes in a common duct the calculation of the three equivalent resistances is quite simple, as illustrated by fig. 3-5. The network transformation shown in this example resembles transformations well-known from electric network theory.

The configuration of coaxial pipes, also shown in fig. 3-4, is not offered on the DH pipeline market. It is, however, interesting from a theoretical point of view, since it provides the possibility of reducing the total heat loss significantly. The heat loss from the supply line to the environment is seen to vanish, i.e. $R'_{10} = \infty$. R'_{20} is given by elementary heat transfer theory. R'_{12} may be calculated from the formula [3-5]:

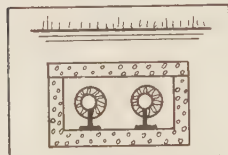
$$R'_{20} = \frac{\frac{\lambda_e}{\lambda_i} \ln \frac{d_o}{d_i} + \operatorname{arcosh} \frac{2h}{d_o}}{2\pi\lambda_e} \quad (3-24)$$



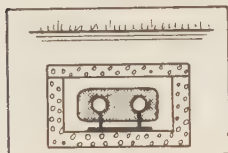
A Separately insulated pipes with separate outer plastic shells.



B Pipes in a common insulation in a common plastic shell.



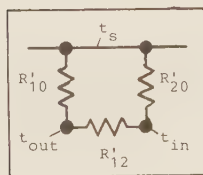
C Separately insulated pipes in a common concrete duct.



D Pipes in a common insulation in a common concrete duct.



E Concentric pipeline design with supply pipe at the centre.



Simple theoretical network model for the heat flow situation at two pipelines underground.

Applicable to the various types of design showed above.

Fig. 3-4 Various types of DH pipeline design and simple equivalent network model.

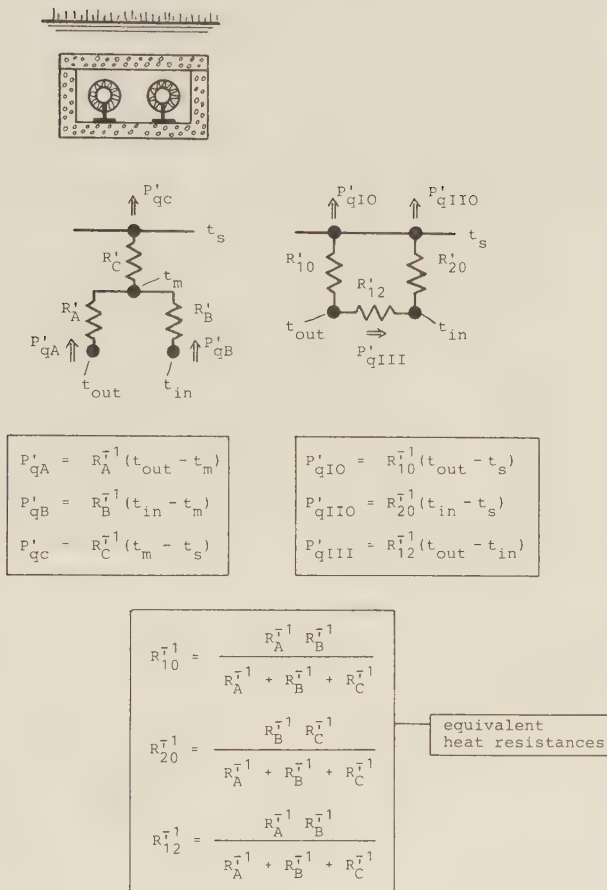


Fig. 3-5

Equivalent heat flow resistances R'_{10} , R'_{20} , and R'_{12} in theoretical network model shown at the bottom of fig. 3-4, applied to the design case of separately insulated pipes underground.

where λ_e and λ_i are heat conductivities of earth and insulation material, respectively. h is the depth of the pipe centre below ground level, and d_o and d_i are outer and inner diameters of the insulation round the return pipe.

The equality $c_{12} = c_{21}$ introduced above, which may be designated as a reciprocity condition, may seem intuitively very natural. However, the strict proof of its general validity to a complex heat conductivity case with two pipes underground requires the aid of mathematical vector theory [3-10].

All symmetrical geometrical configurations are readily seen to fulfil the reciprocity condition. By carrying out experiments with electrically conducting paper, Vidal [3-11] derived values of heat flows influence coefficients for the case of two pipes in a common insulation underground. His results are seen to be in accordance with the reciprocity condition, regardless of the relative sizes of the pipes. Vidal also stated explicitly that the condition holds for any configuration with a homogenous heat-conducting medium, i.e. with pipes having no insulation.

So far we have been concerned with heat flows per unit length of pipes. The total temperature changes in the two pipes from heat generation to the consumer may be found by solving the following pair of differential equations:

$$\frac{dt_{out}}{dx} = \frac{1}{mc} \left[R'_{10} (t_{out} - t_s) + R'_{12} (t_{out} - t_{in}) \right] - \frac{1}{c_p} \frac{dp}{dx} \quad (3-25)$$

$$\frac{dt_{in}}{dx} = - \frac{1}{mc} \left[R'_{20} (t_{in} - t_s) - R'_{12} (t_{out} - t_{in}) \right] - \frac{1}{c_p} \frac{dp}{dx} \quad (3-26)$$

where for constant pipe diameter dp/dx may be considered constant. Here, x runs from the consumer to the heat generation, cf. fig. 3-6.

From eqs. (3-25) and (3-26) it can be seen that the problem of determining variations of supply and return temperatures is formally a heat-exchanger problem with 3 heat-carrying media, one of them (the soil) having constant temperature.

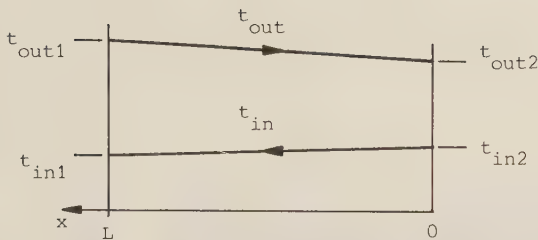


Fig. 3-6

Variable temperatures
in supply and return
pipes.

If t_{out2} and t_{in2} are given, the values of t_{out1} and t_{in1} can be found by putting $x = L$ in the solutions to the pair of differential equations.

Instead of applying this strict kind of procedure we shall introduce linearized equations for the changes in supply and return temperatures with load. This linearization will be acceptable so long as the three heat loss components are small in comparison with the amount of heat transported by the DH system.

Supposing that temperatures t_{out2} , t_{in2} , and t_s , as well as heat flows at reference load conditions are known, we may write down the following equations describing the load performance of the DH system:

$$j = P_{qIO,o}^* / (t_{out2,o} - t_s) \quad (3-27)$$

$$P_{qIO}^* = P_{qIO,o}^* + j(t_{out2} - t_{out2,o}) \quad (3-28)$$

$$k = P_{qII,o}^* / (t_{in2,o} - t_s) \quad (3-29)$$

$$P_{qI,o}^* = P_{qII,o}^* + k(t_{in2} - t_{in2,o}) \quad (3-30)$$

$$l_{10} = P_{qIII,o}^* / (t_{out2,o} - t_{in2,o}) \quad (3-31)$$

$$l_{01} = -l_{01} \quad (3-32)$$

$$P_{qIII}^* = P_{qIII,o}^* + l_{10}(t_{out2} - t_{out2,o}) + l_{01}(t_{in2} - t_{in2,o}) \quad (3-33)$$

$$P_{qc}^* = P_{qIO}^* + P_{qIIO}^* \quad (3-34)$$

$$P_{qc,o}^* = P_{qIO,o}^* + P_{qIIO,o}^* \quad (3-35)$$

$$P_{q1}^* = P_{q2}^* + P_{qc}^* - P_{ec}^* \quad (3-36)$$

$$P_{q1,o}^* = 1 + P_{qc,o}^* - P_{ec,o}^* \quad (3-37)$$

$$t_{in1} = t_{in2} + (t_{out2} - t_{in2}) \frac{y_{ec}^* - P_{qIIO}^* + P_{qIII}^*}{P_{q2}^* - zP_{ec}^*} \quad (3-38)$$

$$t_{in1,o} = t_{in2,o} + (t_{out2,o} - t_{in2,o}) \frac{y_{ec,o}^* - P_{qIIO,o}^* + P_{qIII,o}^*}{1 - zP_{ec,o}^*} \quad (3-39)$$

$$t_{out1} = t_{in1} + (t_{out2} - t_{in2}) \frac{P_{q1}^* + wP_{ec}^*}{P_{q2}^* - zP_{ec}^*} \quad (3-40)$$

$$t_{out1,o} = t_{in1,o} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1,o}^* + wP_{ec,o}^*}{1 - zP_{ec,o}^*} \quad (3-41)$$

The energy flow situation for the DH system is conveniently summarized in the form of a Sankey diagram (see fig. 3-7 [1-24]). Here for graphical reasons heat and pressure losses have been concentrated to a point halfway from heat production to consumption. The total widths of the supply and return line bars represent enthalpy flows of the two pipes. According to the definition of enthalpy, the bars have been divided into internal energy parts (shaded) and a pv-part (blank in the figure). The blank parts are seen to diminish as the water moves round the system from heat generation to heat delivery, via supply and return pipes.

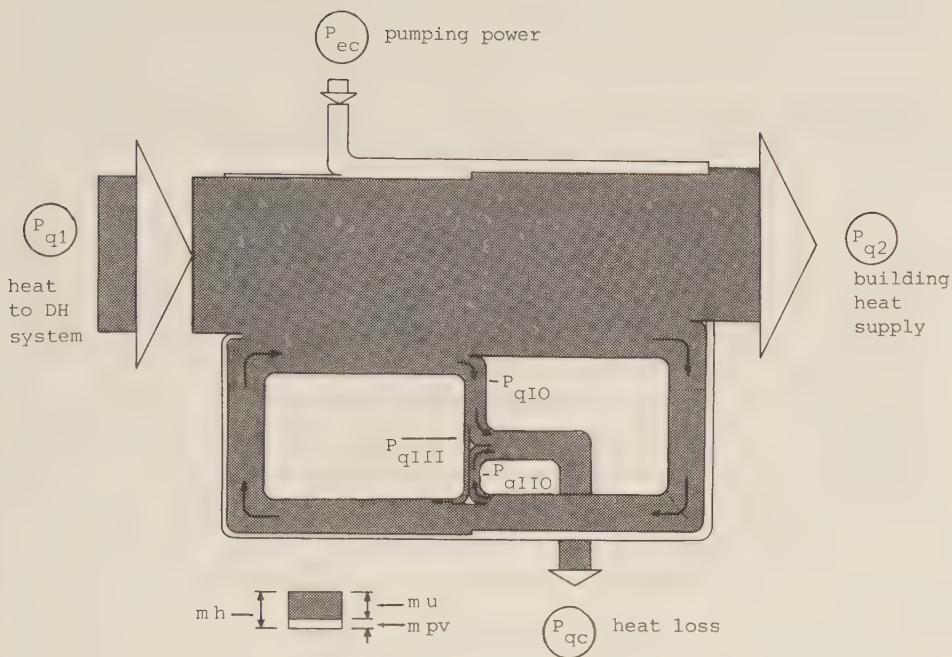


Fig. 3-7

Sankey-diagram for heat flow through DH system.

Heat flow expressed as mass flowrate times enthalpy, where enthalpy is being divided into an energy- and a pv-part.

P_{qIO} = heat flow from supply pipe

P_{qIO} = heat flow from return pipe

P_{qIII} = internal heat flow from supply to return

This kind of Sankey-diagram was presented for the first time in [1-24].

3.3 Heat Exchangers of the District Heating System

In this section we shall derive analytical expressions for the load performances of the two heat exchangers of the district heating system, the condenser of the power plant and the consumer's substation. As the expressions turn out to be analogous for the two types of heat exchangers, we shall concentrate the discussion to one of them (the consumer's substation), and at the end of the chapter analytical expressions will be given for the other heat exchanger.

Analytical solutions for DH supply and return temperatures at different load conditions are not readily given by common expressions characterizing heat exchangers, logarithmic mean temperature difference or heat exchanger effectiveness. Another difficulty is that in the model calculations we are going to let the mass flow of the DH system vary within wide limits, so that in some cases very large pressure drops are developed in the heat exchangers. The effects of such large pressure drops are not taken into account in elementary heat exchanger theory. Therefore below we shall first derive heat exchanger differential equations, then solve them, and make suitable algebraic manipulations with the solutions.

For the heat exchanger of the consumer's substation, in the model calculations in the following chapters, we shall make the simplifying assumption that the temperature change on the secondary side (the medium of the central heating system) is negligible. The significance of this approximation will be discussed below.

The problem of heat transfer from an incompressible fluid with large pressure drop over the heat transfer surface does not seem to have been given much attention in the literature on heat exchangers. A method of handling this phenomenon will therefore be presented in this section.

Case of varying secondary water temperature and of negligible pressure losses

At first we neglect pressure drops according to fig. 3-8. The temperature difference $\Theta(x)$ is defined as:

$$\Theta = t - t' \quad (3-42)$$

The overall heat transfer coefficient is:

$$U_2 = \frac{L}{A_2} \frac{1}{\theta} \frac{dP_{q2}}{dx} \quad (3-46)$$

Eqs. (3-45) and (3-46) give:

$$\frac{d\theta}{dx} = - \frac{U_2^*}{L} \left(1 - \frac{m}{m'}\right) \theta \quad (3-47)$$

where the dimensionless heat transfer coefficient (in Anglo-Saxon literature also designated as the 'Number of Heat Transfer Units', NTU)

$$U_2^* = \frac{U_2 A_2}{mc} \quad (3-48)$$

has been introduced. The solution to eq. (3-47) is, assuming that U_2 is constant with x :

$$\theta = \theta_1 \exp \left(-U_2^* \left(1 - \frac{m}{m'}\right) \frac{x}{L} \right) \quad (3-49)$$

which for $x = L$ gives:

$$\theta_2 = \theta_1 \exp \left(-U_2^* \left(1 - \frac{m}{m'}\right) \right) \quad (3-50)$$

Further:

$$P_{q2} = mc(t_{out2} - t_{in2}) \quad (3-51)$$

together with

$$P_{q2} = m'c(t_1 - t_2) \quad (3-52)$$

gives

$$t_1 - t_2 = \frac{m}{m'} (t_{out2} - t_{in2}) \quad (3-53)$$

Substitutions:

$$t_{out2} - t_{in2} = \theta_1 - \theta_2 + t_1 - t_2 \quad (3-54)$$

$$\theta_1 = \frac{P_{q2}}{mc} \frac{1 - \frac{m}{m'}}{1 - \exp(-U_2^* (1 - \frac{m}{m'}))} \quad (3-55)$$

$$m^* = m/m_{O_1} \quad (3-56)$$

$$m'^* = m'/m'_{O_1} \quad (5-57)$$

$$\mu = m_{O_1}/m'_{O_1} \quad (3-58)$$

give, for $\mu \neq 1$:

$$\frac{t_{out2} - t_1}{t_{out2,O} - t_{1,O}} \frac{1}{P_{q2}^*} = \frac{1}{m^*} \frac{1 - \frac{m^*}{m'^*} \mu}{1 - \mu} \frac{1 - \exp(-U_{2,O}^* (1 - \mu))}{1 - \exp(-U_2^* (1 - \frac{m^*}{m'^*} \mu))} \quad (3-59)$$

The left-hand side of this formula may be regarded as a dimensionless supply water temperature. A corresponding dimensionless return temperature can also be derived by algebraic manipulations. First we write:

$$t_{in2} - t_1 = (t_{in2} - t_2) - (t_1 - t_2) \quad (3-60)$$

For the first of the right-hand terms we have:

$$t_{in2} - t_2 = \theta_2 = \theta_1 \exp(-U_2^* (1 - \frac{m}{m'})) \quad (3-61)$$

Here, θ_1 can be found from:

$$\theta_1 - \theta_2 = (t_{out2} - t_{in2}) - (t_1 - t_2) \quad (3-62)$$

Eqs. (3-50), (3-53), and (3-62) give:

$$\theta_1 = (t_{out2} - t_{in2}) \frac{1 - \frac{m}{m'}}{1 - \exp(-U_2^* (1 - \frac{m}{m'}))} \quad (3-63)$$

which in combination with eqs. (3-50) and (3-51) yields:

$$t_{in2} - t_2 = \frac{P_{q2}}{mc} \frac{1 - \frac{m}{m'}}{\exp(U_2^* (1 - \frac{m}{m'})) - 1} \quad (3-64)$$

For the second of the right-hand terms in eq. (3-60) we get, combining eqs. (3-51) and (3-53):

$$t_1 - t_2 = \frac{m}{m'} \frac{P_{q2}}{mc} \quad (3-65)$$

Insertion of eqs. (3-64) and (3-65) in (3-60) gives:

$$t_{in2} - t_1 = \frac{P_{q2}}{mc} \left[\frac{1 - \frac{m}{m'}}{\exp(U_2^* (1 - \frac{m}{m'})) - 1} - \frac{m}{m'} \right] \quad (3-66)$$

which, combined with eq. (3-55) yields, for $\mu \neq 1$:

$$\frac{t_{in2} - t_1}{t_{out2,o} - t_{1,o}} \frac{1}{P_{q2}^*} = \frac{1}{m^*} \frac{1 - \frac{m^*}{m'^*} \mu \exp(U_2^* (1 - \frac{m^*}{m'^*} \mu))}{1 - \mu} \cdot \frac{1 - \exp(-U_{2,o}^* (1 - \mu))}{\exp(U_2^* (1 - \frac{m^*}{m'^*} \mu)) - 1} \quad (3-67)$$

On the right-hand sides of both eqs. (3-59) and (3-67) we have the dimensionless heat transfer coefficient, U_2^* , which in an off-design case in general will deviate from its reference value, $U_{2,o}^*$.

Before proceeding we shall assume that heat transfer resistances in the heat exchanger wall and in possible deposit layers, can be neglected, so

that the overall heat transfer coefficient may be written as:

$$U_2 = (1/U_{2A} + 1/U_{2B})^{-1} \quad (3-68)$$

where U_{2A} and U_{2B} are heat transfer coefficients of the primary and secondary sides respectively. For the ratio of the two coefficients at reference load we introduce the designation:

$$x_2 = U_{2A,o}/U_{2B,o} \quad (3-69)$$

For both U_{2A} and U_{2B} a dimensionless equation of the form:

$$Nu = \text{const. } Re^p Pr^q \quad (3-70)$$

will be valid, so that we may write, neglecting temperature influence on the heat transfer process:

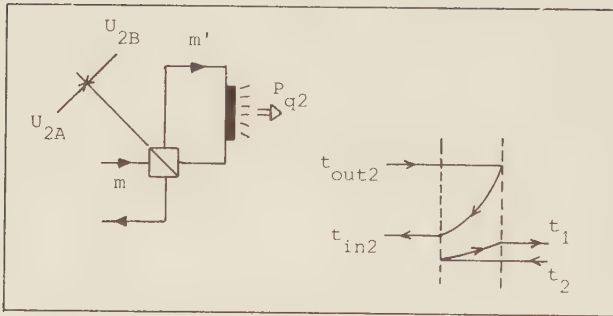
$$U_{2A} = U_{2A,o} m^{*p_{2A}} \quad (3-71)$$

$$U_{2B} = U_{2B,o} m'^{*p_{2B}} \quad (3-72)$$

The assumption of no temperature influence on heat transfer coefficients represents a simplification of our calculations. In the Nusselt-, Reynolds-, and Prandtl Numbers four physical properties of water appear, namely: the conductivity of heat, the density, the specific heat and the dynamic viscosity. Of these the first three vary only a little with temperature, while the variation of the dynamic viscosity is significant. However, in the Reynolds Number it appears in the denominator, and in the Prandtl Number in the nominator, so that the resulting temperature influence is not very dramatic. For instance, a (great) temperature variation from 50°C to 150°C will give an improvement of heat transfer of around 40%.

From eqs. (3-69), (3-70), (3-71), and (3-72) we have:

$$\frac{U_2}{U_{2,o}} = \frac{1 + x_2}{m^{*p_{2A}} + x_2 m'^{*p_{2B}}} \quad (3-73)$$



$$\begin{aligned}
 m' &= m'_o \text{ (const.)} \\
 U_{2,o}^* &= 1.5 \\
 p_2 &= 0.5 \\
 x_2 &= 1.0 \\
 &= U_{2A,o}/U_{2B,o}
 \end{aligned}$$

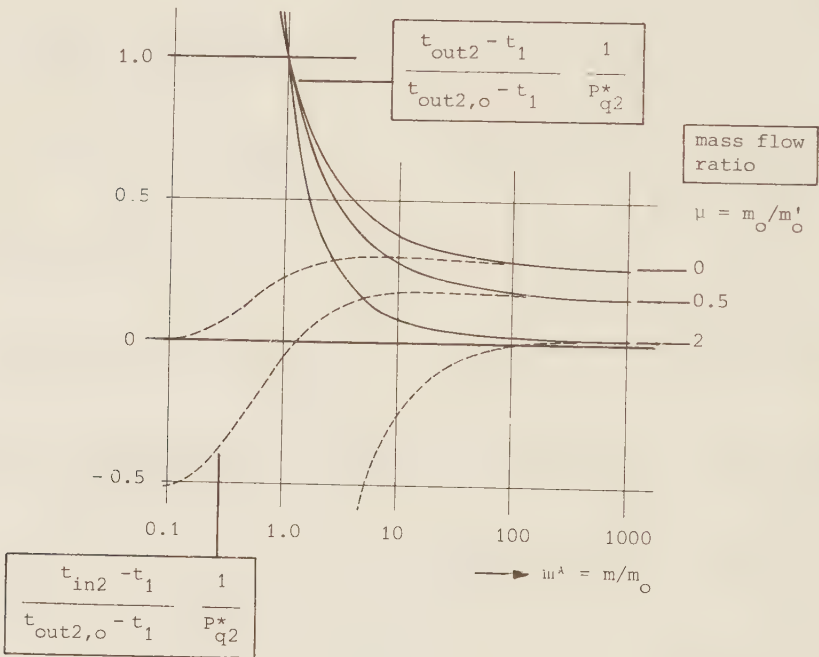


Fig. 3-9 Dimensionless DH supply and return temperatures for varying operational conditions of consumer's heat exchanger.

Case of secondary constant secondary water flow-rate and of negligible pressure losses.

Logarithmic scale on the abscissa.

According to the definition of U_2^* :

$$U_2^* = \frac{U_{2,A} A_2}{mc} = \frac{U_{2,o} A_2}{m_o c} \frac{U_2 / U_{2,o}}{m^*} = \frac{U_{2,o}^*}{m^*} \frac{U_2}{U_{2,o}} \quad (3-74)$$

which, combined with eq. (3-73), gives:

$$U_2^* = \frac{U_{2,o}^*}{m^*} \frac{1 + x_2}{m^* \frac{-P_{2A}}{m^*} + x_2 \frac{-P_{2B}}{m'^*}} \quad (3-75)$$

In the special case of $m'^* = 1$, i.e. constant flow on the secondary side, U_2^* and the dimensionless temperatures according to eqs. (3-59) and (3-67) become functions of m^* only, for different operating conditions.

Fig. 3-9 is a plot of those functions, with the further specialization of $x_2 = 1.0$, and with the ratio $\mu = m_o / m'_o$ as curve parameter.

The logarithmic abscissa has here been extended to very large values of m^* . Because of pressure losses, so far neglected in the calculations, such values of m^* are of more theoretical than practical interest.

For $P_{q2}^* = \text{constant}$ and $\mu = \text{constant}$, following one pair of curves, we see that the curve for $t_{\text{out}2}$ always goes down with increasing m^* , i.e. the lower the supply temperature, the larger a primary flow is needed to keep a constant heat flow across the heat exchanger. At lower values of m^* , the dimensionless return temperature rises with m^* . At higher values of m^* , the variation is small; for $\mu = 0$ and $\mu = 0.5$, a weak maximum of the curves is seen. The larger μ is, the more lively is the variation of the return temperature, and the lower is the lower limit values of $t_{\text{out}2}$ at large primary flows.

The decrease of $t_{\text{out}2}$ with increasing m^* is partly a result of an improved U_2 . However, with increasing m^* the marginal increase in U_2 goes down, since U_2 becomes dependable to an increasing degree upon U_{2B} (the heat transfer coefficient on the secondary side), which is unaffected by m^* . The drop in $t_{\text{out}2}$ at lower values of m^* may also in part be interpreted as a consequence of the rise in $t_{\text{in}2}$. This may be seen from the expression:

$$P_{q2} = U_2 A_2 \theta_m \quad (3-76)$$

with the logarithmic mean temperature difference:

$$\theta_m = \frac{(t_{out2} - t_1) - (t_{in2} - t_2)}{\ln \frac{t_{out2} - t_1}{t_{in2} - t_2}} \quad (3-77)$$

Even if there were no improvement of U_2 with increasing m^* , P_{q2} could still be kept constant, namely if t_{out2} decreases and t_{in2} increases so that θ_m is kept constant.

Case of constant secondary water temperature and of negligible pressure losses

Eqs. (3-59) and (3-67) for supply and return temperatures in this case simplify to:

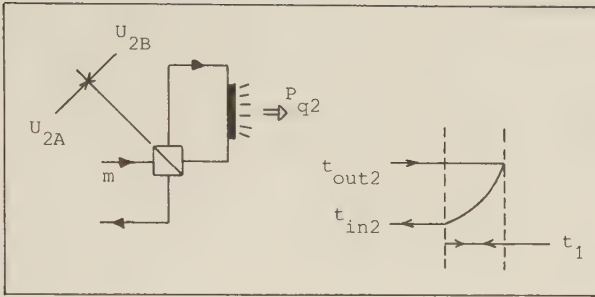
$$\frac{t_{out2} - t_1}{t_{out2,o} - t_{1,o}} \frac{1}{P_{q2}^*} = \frac{1}{m^*} \frac{1 - \exp(-U_{2,o}^*)}{1 - \exp(-U_2^*)} \quad (3-78)$$

and

$$\frac{t_{in2} - t_1}{t_{out2,o} - t_{1,o}} \frac{1}{P_{q2}^*} = \frac{1}{m^*} \frac{1 - \exp(-U_{2,o}^*)}{\exp U_2^* - 1} \quad (3-79)$$

In fig. 3-10 these two functions have been plotted for three values of the ratio x_2 between heat transfer coefficients. As in fig. 3-9 the heat transfer coefficient U_{2B} of the secondary side has been assumed to be constant. It is seen that for $x_2 = 0.1$, there is a quite distinct maximum of the return temperature curve.

The limit values of the dimensionless supply and return temperatures at high values of m^* can be found with the help of l'Hôpital's rule. Substituting $s = 1/m^*$ in the right-hand sides of eqs. (3-78) and (3-79) gives fractions with vanishing nominators and denominators for $s = 0$. Differenti-



$$\begin{aligned} U_{2,o}^* &= 1.5 \\ P_2 &= 0.5 \end{aligned}$$

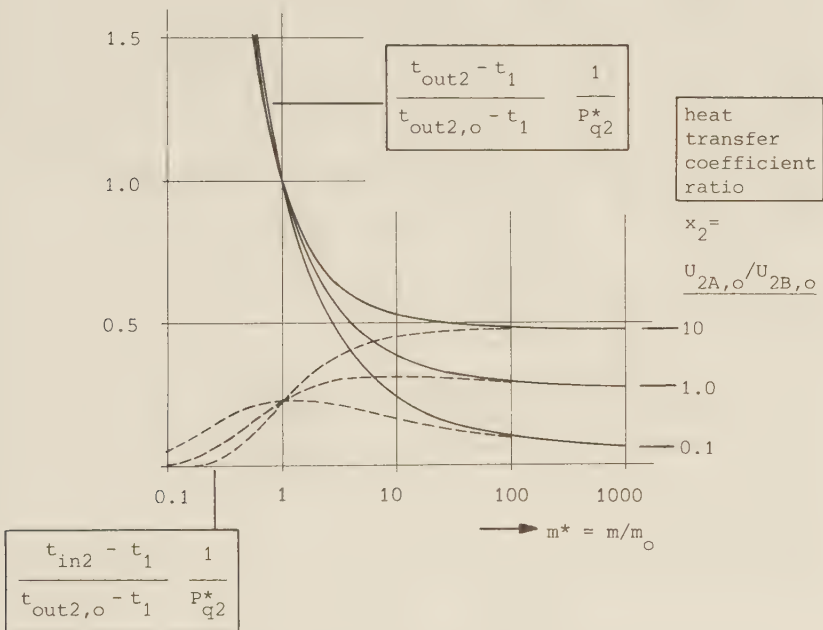


Fig. 3-10 Dimensionless DH supply and return temperatures for varying operational conditions of consumer's heat exchanger.

Case of constant secondary water temperature and of negligible pressure losses.

Logarithmic scale on the abscissa.

ating the nominators and denominators and putting $s = 0$ then gives (omitting intermediate calculations):

$$\begin{aligned} \lim_{m^* \rightarrow \infty} \left[\frac{t_{\text{out}2} - t_1}{t_{\text{out}2,o} - t_{1,o}} \frac{1}{P_{q2}^*} \right] &= \lim_{s \rightarrow 0} \left[\frac{t_{\text{out}2} - t_1}{t_{\text{out}2,o} - t_{1,o}} \frac{1}{P_{q2}^*} \right] = \\ &= \frac{x_2}{1 + x_2} \frac{1 - \exp(-U_{2,o}^*)}{U_{2,o}^*} \end{aligned} \quad (3-80)$$

$$\begin{aligned} \lim_{m^* \rightarrow \infty} \left[\frac{t_{\text{in}} - t_1}{t_{\text{out}2,o} - t_{1,o}} \frac{1}{P_{q2}^*} \right] &= \lim_{s \rightarrow 0} \left[\frac{t_{\text{in}} - t_1}{t_{\text{out}2,o} - t_{1,o}} \frac{1}{P_{q2}^*} \right] = \\ &= \frac{x_2}{1 + x_2} \frac{1 - \exp(-U_{2,o}^*)}{U_{2,o}^*} \end{aligned} \quad (3-81)$$

The equality of the two limit temperatures is a consequence of neglecting pressure losses which, for constant P_{q2}^* means that

$$\lim_{m^* \rightarrow \infty} (t_{\text{out}2} - t_{\text{in}2}) = 0 \quad (3-82)$$

The limiting values of supply and return temperatures for great primary mass flowrates are seen to go down with decreasing values of the ratio x_2 , and in the theoretical case of $x_2 = 0$, the limiting values of the primary side temperatures become equal to the secondary water temperature. The explanation is that with $x_2 = 0$ all heat transfer resistance is concentrated to the primary side of the heat exchanger, and at 'infinitely' large values of m^* this resistance vanishes.

At high values of m^* in fig. 3-10, further increases in m^* give only small marginal decreases of the supply temperature. In such cases it would be more beneficial to increase the secondary mass flow instead. In practice, greatly differing values of heat transfer coefficients on the two sides of the heat transfer surface, would represent an unfavourable allocation of pressure losses between the two sides.

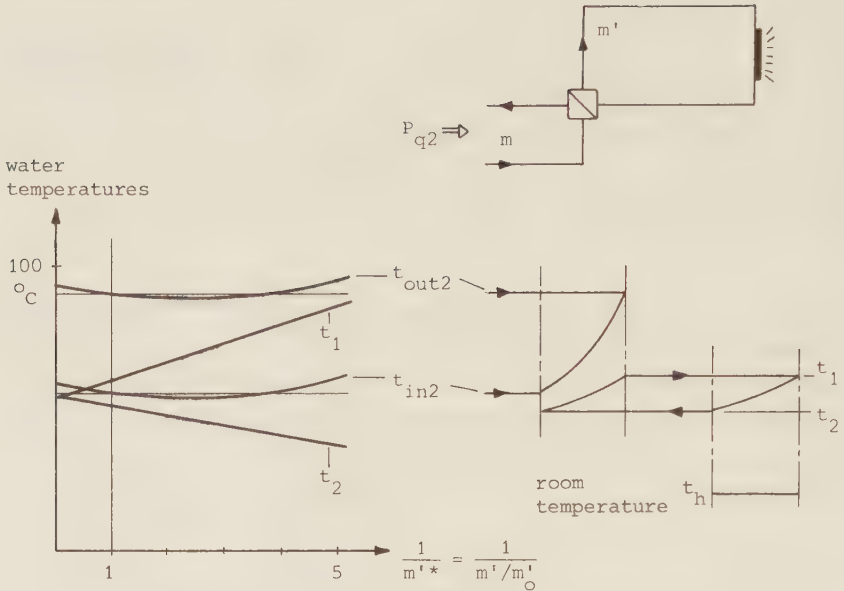
In the development of equations leading to figs. 3-9 and 3-10, it was supposed that heat transfer resistance in the wall of the heat exchanger was negligible. Practical designs are usually based on so small a wall thickness that the wall resistance will not dominate over boundary layer resistances. However, fouling may lead to a build-up of layers with significant heat transfer resistance. Since in both the two figures the mass flowrate of the secondary side is kept constant, the diagrams could be interpreted to include cases of heat transfer resistance in wall or deposit layers, by including those resistances in the value for U_{2B} , the heat transfer resistance of the secondary side.

In the model calculations in later chapters we are going to adopt the simplification of a constant secondary water temperature. By comparing figs. 3-9 and 3-10 we see that as a consequence of this simplification, we shall have less lively variations in DH return temperature than in systems from practice. Typically, secondary side temperature differences may amount to 20°C at maximum heat load, and to smaller differences at lower loads.

Fig. 3-11 illustrates some effects of varying the secondary mass flowrate for a given radiator system. Here the primary flowrate is kept constant, and, for simplicity, the overall heat transfer coefficient is also supposed to be constant at variations in secondary flowrate. The abscissa is the relative value of the inverse of the secondary flowrate. It can be seen that there is a minimum of DH supply temperature at a certain value of the abscissa. If instead of the primary mass flowrate the DH supply temperature had been kept constant, we can expect the DH return temperature to have a minimum at a certain secondary flowrate. Winkens and his co-workers [3-12], [1-16], have investigated how a lowering of DH return temperature may be achieved by adopting unconventionally small flowrates in building radiator systems.

Case of constant secondary water temperature and of varying primary side pressure losses

The calculations have so far been made without taking pressure losses into account. Since the mass flow of the secondary side is kept constant, it can be justified to avoid the analytic complications of the usually small pressure losses on the secondary side. But for primary side pressure losses a different situation exists. In figs. 3-9 and 3-10 we have examined opera-



abscissa: the inverse of the dimensionless water flowrate in the radiator system

Fig. 3-11

Effect on water temperatures of varying the flowrate in the radiator system.

Simplifying assumptions made:

- constant overall heat transfer coefficient in consumer's heat exchanger,
- constant mass flow m in DH system,
- negligible pressure losses in heat exchanger,
- constant building heat supply P_{q2} .

tional conditions with DH mass flows several decades greater than the reference value. The pumping power needed to overcome heat exchanger pressure losses increases with the cube of the mass flow, so that even with a small pumping power at reference mass flow, it soon becomes of considerable magnitude when m^* is increased.

In practice a very large pumping power will not be accepted because of accompanying erosion, noise, and other technological problems, as discussed in section 4.2. However, it would be unsatisfactory from a theoretical point of view not to consider pressure losses in heat exchangers.

Standard textbooks on heat transfer deal with problems of high fluid velocity [3-13], primarily as a result of the need to calculate accurately the thermal loading of aeroplanes or space shuttles, travelling at high speed in the atmosphere. Pohlhausen obtained solutions for the temperature distribution in the boundary layer by introducing the concept of a recovery factor:

$$r = \frac{T_{aw} - T_{\infty}}{T_T - T_{\infty}} \quad (3-83)$$

where T_{∞} is the fluid temperature outside the boundary layer.

$$\Delta T = T_T - T_{\infty} = \frac{v_{\infty}^2}{2c_p} \quad (3-84)$$

is the dynamic temperature of free fluid with the relative velocity v_{∞} , and T_{aw} is the actual, adiabatic wall temperature. I.e., the recovery factor is the ratio of actual temperature rise and isentropic temperature rise of the free fluid. However, this theory applies to compressible fluids, and in the present DH application we have the incompressible fluid of water. In a full treatment of this slightly different case, it would be necessary to formulate boundary layer equations and solve them. Even in the idealized case of a flat plate this is not quite simple, and with more complicated geometric configurations the analytical difficulties increase. Instead, we shall adopt here an integral, semianalytical approach, retaining the concept of a recovery factor, but in a sense somewhat different than for r , defined above for a compressible fluid.

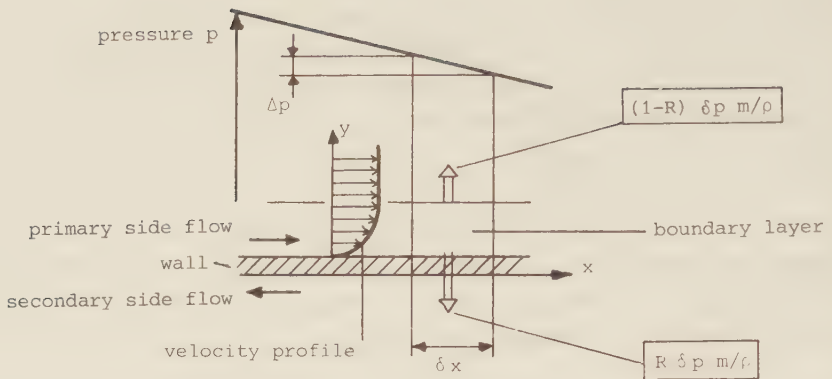


Fig. 3-12 Recovery factor R referring to incompressible fluid flow developing great pressure drops at heat exchanger surface.

As shown in fig. 3-12, the amount of heat

$$\delta p \frac{m}{\rho}$$

developed by the pressure drop δp over a small length δx of the heat transfer surface is thought to be split into two parts:

$R \delta p \frac{m}{\rho}$ being developed directly on the heat transfer surface, and

$(1-R) \delta p \frac{m}{\rho}$ being developed in the main fluid, outside the boundary layer.

For the first of these two heat parts, the boundary layer represents no thermal transmission resistance at all, whereas the second part of the heat adds to the heat stored in the main fluid, a heat which is transmitted to the wall surface with the full transfer resistance of the boundary layer.

This way of splitting up the heat generated in the boundary layer represents a discretization of a continuous physical reality. Here, at each height, y , over the wall surface is generated a differential friction heat

to be transferred partly to the wall, and partly to the bulk fluid, in fractions depending upon the variable height, y .

Selecting a value for the recovery factor R , means settling how close to the wall the 'mean' frictional heat is considered to be generated. In the following we shall for simplicity assume that R is independent of x .

Having introduced R , we may pass from the finite element description of fig. 3-12 to a differential equation for the temperature difference θ between primary and secondary water flows (cf. fig. 3-8):

$$\frac{d\theta}{dx} = (1 - R) \frac{\Delta p/L}{\rho c} - U_2^* \frac{1}{L} \theta \quad (3-85)$$

with:

$$U_2^* = \frac{U_2 A_2}{m c} \quad (3-86)$$

Eq. (3-85) is analogous to eq. (3-49), valid for the case of vanishing pressure losses.

The solution to eq. (3-85) is:

$$\theta = (\theta_1 - \theta_p) \exp \left(-U_2^* \frac{x}{L} \right) + \theta_p \quad (3-87)$$

where the constant θ_p is:

$$\theta_p = (1 - R) \frac{\Delta p}{\rho c} \frac{1}{U_2^*} \quad (3-88)$$

Putting $x = L$ gives:

$$\theta_2 = (\theta_1 - \theta_p) \exp (-U_2^*) + \theta_p \quad (3-89)$$

Denoting the total pumping power of the DH system P_{ec} (as in the preceding section), and z the fraction of this power needed to overcome primary side pressure losses in the consumer substation heat exchanger, we have

$$zP_{ec} = \frac{m}{\rho} \Delta p \quad (3-90)$$

At reference load the heat transmitted to the heat exchanger wall is:

$$P_{q2,o} = m_o c (t_{out2,o} - t_{in2,o}) + zP_{ec,o} \quad (3-91)$$

Assuming that $zP_{ec,o}$ is so small that it can be neglected, we get by dividing eq. (3-90) by eq. (3-91):

$$zP_{ec}^* = \frac{m^*}{\rho c} \frac{\Delta p}{t_{out2,o} - t_{in2,o}} \quad (3-92)$$

z may be regarded as a constant at different load conditions, and since Δp varies proportionally to the cube of m^* , we get

$$zP_{ec}^* = m^{*3} zP_{ec,o}^* \quad (3-93)$$

which in combination with eqs. (3-90) and (3-92) gives:

$$\theta_p = (1 - R) \frac{m^{*2}}{U_2^*} zP_{ec,o}^* (t_{out2,o} - t_{in2,o}) \quad (3-94)$$

In the model calculations in later chapters numerical values are going to be specified for $t_{out2,o}$ and $t_{1,o}$, but not $t_{in2,o}$. We therefore in eq. (3-94) make the approximation:

$$t_{out2,o} - t_{in2,o} = (t_{out2,o} - t_{1,o}) (1 - \exp(-U_{2,o}^*)) \quad (3-95)$$

(a good approximation, since $\theta_{p,o}$ is small compared with $t_{out2,o} - t_{in2,o}$), so that:

$$\theta_p = (1-R) \frac{m^{*2}}{U_2^*} zP_{ec,o}^* (t_{out2,o} - t_{1,o}) (1 - \exp(-U_{2,o}^*)) \quad (3-96)$$

The equation:

$$P_{q2} = mc(t_{out2} - t_{in2}) + zP_{ec} \quad (3-97)$$

together with:

$$P_{q2,o} = m_o c(t_{out2,o} - t_{in2,o}) + zP_{ec,o} \quad (3-98)$$

gives:

$$\frac{t_{out2} - t_{in2}}{t_{out2,o} - t_{in2,o}} = \frac{1}{m^*} \frac{P_{q2}^* - zP_{ec}^*}{1 - zP_{ec,o}^*} \quad (3-99)$$

With $\theta_{p,o} \approx 0$ we get the approximate formula for the supply temperature:

$$\frac{t_{out2} - t_1}{t_{out2,o} - t_{1,o}} = \frac{\theta_p}{t_{out2,o} - t_{1,o}} + \frac{P_{q2}^* - zP_{ec}^*}{1 - zP_{ec,o}^*} \frac{1}{m^*} \frac{1 - \exp(-U_{2,o}^*)}{1 - \exp(-U_2^*)} \quad (3-100)$$

where θ_p can be calculated from eq. (3-96).

For the return temperature we analogously get

$$\frac{t_{in2} - t_1}{t_{out2,o} - t_{1,o}} = \frac{\theta_p}{t_{out2,o} - t_{1,o}} + \frac{P_{q2}^* - zP_{ec}^*}{1 - zP_{ec,o}^*} \frac{1}{m^*} \frac{1 - \exp(-U_{2,o}^*)}{1 - \exp(-U_2^*)} \exp(-U_2^*) \quad (3-101)$$

Assuming that the flow on the secondary side is constant, we have from eq. (3-75):

$$U_2^* = \frac{U_{2,o}^*}{m^*} \frac{1 + x_2}{m^* 2A + x_2} \quad (3-102)$$

The temperature drop on the primary side may be found by subtracting eq. (3-101) from eq. (3-100):

$$\frac{t_{out2} - t_{in2}}{t_{out2,o} - t_{1,o}} = \left(\frac{t_{out2} - t_1}{t_{out2,o} - t_{1,o}} - \frac{\theta_p}{t_{out2,o} - t_{1,o}} \right) (1 - \exp(-U_2^*)) \quad (3-103)$$

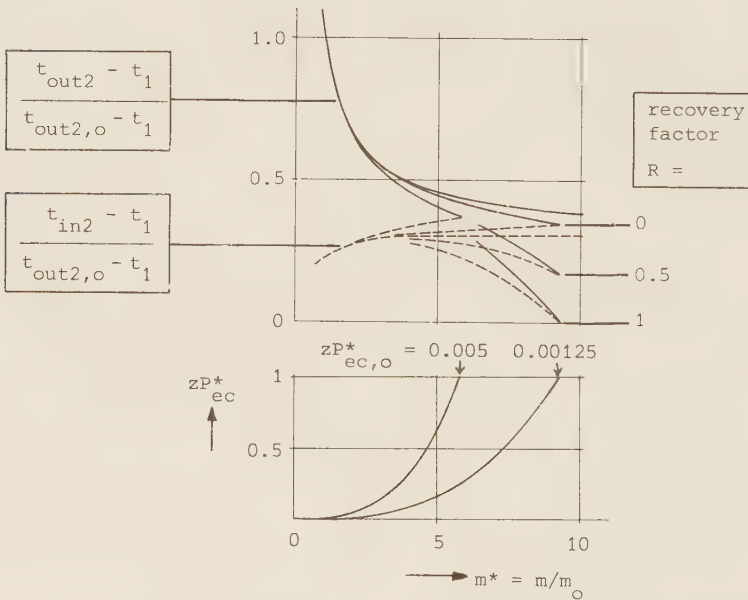
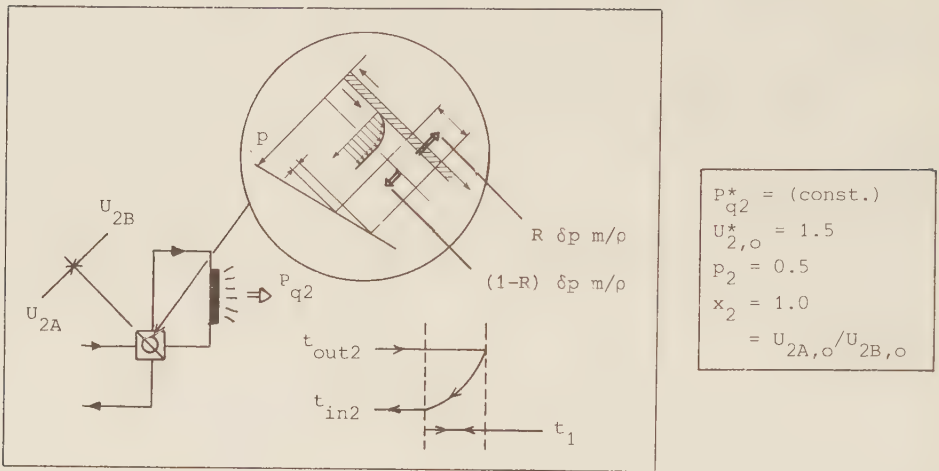


Fig. 3-13 Dimensionless DH supply and return temperatures for varying operational conditions of consumer's heat exchanger.

Case of constant secondary water temperature and of varying primary side pressure losses.

In fig. 3-13 the dimensionless supply and return temperatures according to eqs. (3-100) and (3-101) have been plotted for some combinations of numerical constants. Note that in opposition to figs. 3-9 and 3-10, the abscissa here is linear, and that the inverse of P_{q2}^* is not included in the denominators of the expressions for the dimensionless temperatures. At the bottom of fig. 3-13 is plotted the pumping power, zP_{ec}^* , dissipated in the heat exchanger, as a function of different values of m^* .

Two parameters have been varied in the figure:

- The pumping power, $zP_{ec,o}^*$ at reference load, and
- The recovery factor R defined above.

As the bottom curves show, zP_{ez}^* , though initially (at $m^* = 1$) very small, rapidly increases. It soon reaches the value 1, i.e. the extreme case when the pumping power alone is sufficient to develop the heat required by the internal heating system of the building. At the corresponding value of m^* the curves for the supply and return temperatures meet, i.e. there is no temperature drop over the heat exchanges on the primary side. When $zP_{ec,o}^* = 0$, the two curves are seen to approach each other with increasing m^* , but they never meet.

Comparing cases of different values for R in fig. 3-13, we see that the larger R is the more the curves bend downward at high values of m^* . In the extreme case of $R = 1$ and $zP_{ec}^* = 1$ the curves meet on the abscissa; i.e. all terminal temperatures of the heat exchanger are identical, and all the heat generated by primary side pressure drop is transmitted to the secondary side without any driving temperature difference across the heat exchanger.

To simplify the model calculations in the later chapters, we shall assume that

$$R = 1 \quad (3-104)$$

It is clear that the exact value must lie somewhere between 0 and 1, but without a deeper analysis it is difficult to make any certain estimates.

In any case, as long as $zP_{ec}^* \ll 1$, the numerical consequences of the approximation made are not great. If R is given too high a value this results in an overestimation of the heat transfer capacity of the heat exchanger. Thus, the resulting error could at least be in part compensated by adjusting the heat transfer coefficient downwards. Another constant which may be selected to give the best possible analytical description of the heat transfer over a wide range of mass flows, is the exponent p_{2A} in the equation:

$$U_{2A} = U_{2A,o} m^* p_{2A} \quad (3-105)$$

for the primary side heat transfer.

In this basically empirical relation, p_{2A} is normally treated as a constant. This assumption may, however, be rather crude when the range of variation for m^* is large.

To sum up, a stringent treatment of the heat transfer at small and large pressure drops is extremely complex and calls for further basic research. The approach adopted here, based on the concept of a factor R , is approximate, but it makes simplified model calculations possible.

Putting $R = 1$ and thus $\theta_p = 0$ in eqs. (3-100) and (3-103) gives:

$$t_{out2} = t_1 + (t_{out2,o} - t_{1,o}) \frac{\frac{p_{q2}^* - zP_{ec}^*}{1 - zP_{ec,o}^*}}{m^*} \frac{1}{1 - \exp(-U_{2,o}^*)} \quad (3-106)$$

and

$$t_{in2} = t_{out2} - (t_{out2} - t_1)(1 - \exp(-U_2^*)) \quad (3-107)$$

Eqs. (3-106) and (3-107) will be utilized in our model calculations.

For the condensor case we shall assume that pressure losses occur on the DH side alone (here the reader may be referred to the discussion in the next section on the back-pressure turbine), and that $R = 0$, so that the heat w_{ec}^P ,

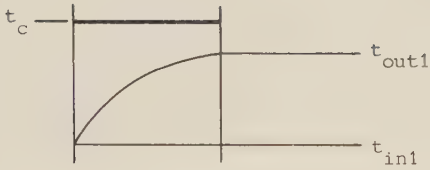


Fig. 3-14

Water heating in turbine condenser.

generated on the DH side, is carried over to the water directly, without passing the heat transfer resistance of the boundary layer. In that case:

$$t_c - t_{out1} = (t_c - t_{in1}) \exp(-U_1^*) \quad (3-108)$$

with

$$U_1^* = \frac{U_1 A_1}{mc} \quad (3-109)$$

giving

$$t_{out1} - t_{in1} = (t_c - t_{in1}) (1 - \exp(-U_1^*)) \quad (3-110)$$

For the heat added to the water in the condensor we have:

$$P_{q1} = mc(t_{out1} - t_{in1}) - wP_{ec} \quad (3-111)$$

which with the help of the relation:

$$P_{q2,o} = m_o c(t_{out2,o} - t_{in2,o}) + zP_{ec,o} \quad (3-112)$$

gives

$$\frac{t_{out1} - t_{in1}}{t_{out2,o} - t_{in2,o}} = \frac{1}{m^*} \frac{p^*_{q1} + w p^*_{ec}}{1 - p^*_{ec,o}} \quad (3-113)$$

By combining eqs. (3-110) and (3-113) we get:

$$t_c = t_{in1} + \frac{1}{m^*} \frac{p^*_{q1} + w p^*_{ec}}{1 - z p^*_{ec,o}} \frac{t_{out2,o} - t_{in2,o}}{1 - \exp(-U^*_1)} \quad (3-114)$$

which will be utilized in the model calculations.

3.4 The Back-Pressure Steam Cycle

The electric output of the back-pressure turbine is dictated by the DH load. The heat requirement of the network is decisive for the amount of steam passing the condenser, and thereby passing the turbine. The DH supply and return temperature levels (in the case of single-stage heating of DH water primarily the supply temperature) influence the steam-side pressure, and thus the enthalpy drop over the turbine.

Fig. 3-15 shows the steam cycle in an enthalpy-entropy Mollier diagram. The line of condensation (4-1) is assumed to follow an isobar, i.e. we neglect the usually small steam-side pressure drop over the condenser. This simplification is made with some hesitation, since in our model calculation we are careful to take into account water side pressure drops in the condenser. As heat transfer by steam condensation is more effective than heat transfer by forced convection of water, it is possible to design the condenser for a smaller pressure drop on the steam side, compared with the water side. On the other hand, the specific volume of steam is much greater than that of water, so that it is by no means evident that the pumping power needed to overcome the water side pressure drop is always smaller than the loss in turbine shaft work from steam side pressure drop. However, steam side pressure drops are usually in the order of no more than 0.005 to 0.5 in. Hg [3-15], so that they will not normally be responsible for great losses in electric output.

Another type of simplification is represented by the discussion of the steam cycle in terms of static enthalpies, when total enthalpies (including velocity-terms) give the full picture of the energy exchanges in the different parts of the cycle. This simplification is primarily of numerical importance for the endpoint of the turbine expansion (point 4 in fig. 3-15). With condensation turbines the end-velocity losses are known to influence the turbine shaftwork significantly, in particular when the cooling-water temperature is low. For instance, with a steam exit velocity of 300 m/sec, the end-velocity loss is 45 kJ/kg, which may be compared to a typical enthalpy drop of 1000 kJ/kg over a turbine. Extrapolating to back-pressure turbines, one might at first expect the end-velocity losses to be vanishing, since the temperature level in that case is much higher, with corresponding smaller specific steam-volume. However, this way of reasoning is misleading. In fact,

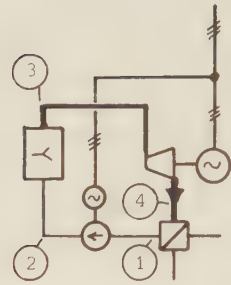
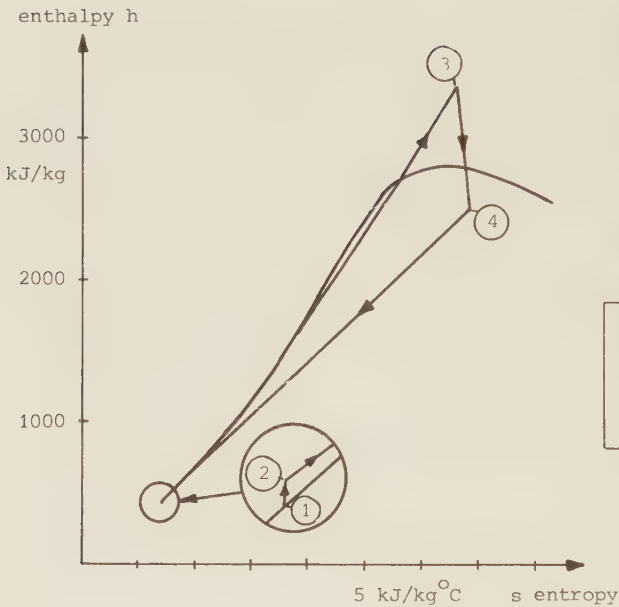


Fig. 3-15

Back-pressure steam cycle (above) shown in Mollier diagram (left).

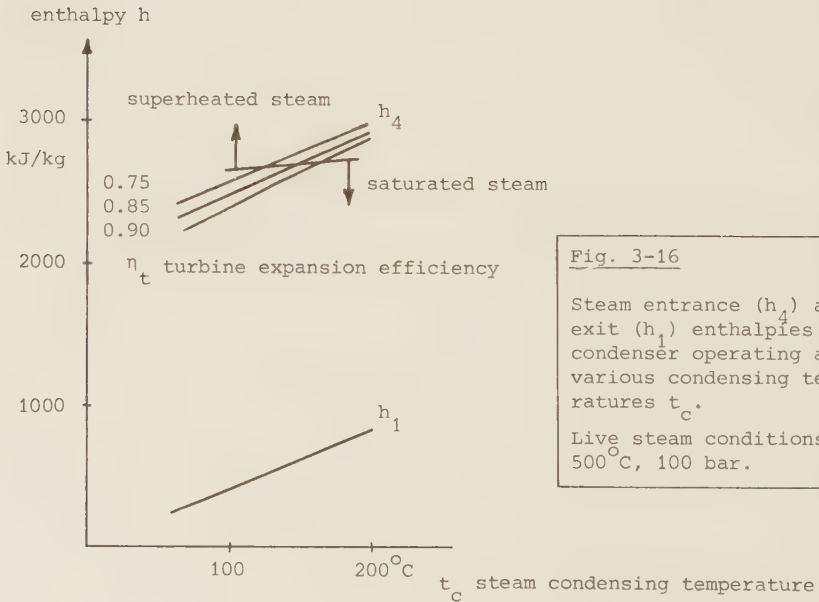


Fig. 3-16

Steam entrance (h_4) and exit (h_1) enthalpies for condenser operating at various condensing temperatures t_c .

Live steam conditions: 500°C, 100 bar.

from turbine theory it is known that the shaft work of a stage, operating not too far from its efficiency optimum, is closely tied to the peripheral speed of the blades. This means that although it would be possible to reduce end-velocity losses by selecting a relatively large exhaust area for the steam, this would also reduce the shaft work of the last stage. Thus, if the number of turbine stages is to be kept low, significant steam exit velocities are difficult to avoid, irrespective of the temperature level of the steam. Part of the end-velocity losses may be recovered in an exhaust diffuser, as is the practice with condensing turbines. With DH back-pressure turbines, the incitement to include this design feature remains.

Furthermore we shall assume that subcooling of condensate will not occur, i.e. in the Mollier diagram point 1 will always lie on the saturation line. Subcooling is unwanted, as it increases the amount of heat delivered in the condenser by each unit of steam, which means that for a given total heat requirement from the DH network the steam flow to the condenser is reduced. Provided maximum steam consumption of the turbine has not been reached, its output is then reduced accordingly.

When in the following different operational conditions for the steam cycle are considered, we shall assume that the admission point of the turbine (point 3 in fig. 3-15) is fixed. This corresponds to nozzle-governing of the turbine, commonly found in practice, although variable admission pressure control is also possible. Sometimes plant control is designed to keep the pressure in the drum of a boiler constant, so that the turbine admission pressure will vary moderately, because of variations in pressure drop in high-temperature pipes.

Fig. 3-16 shows changes in enthalpies of turbine end-expansion, h_4 , and in enthalpies of condensed water, h_1 . Here, expansion takes place from a fixed admission condition with different turbine efficiencies, resulting in the family of curves shown. A limit curve in the figure indicates where the incoming steam to the condenser is saturated, and where it is superheated. In the latter case the abscissa, t_c , is the temperature of the condensing steam and not the superheated steam temperature. The heat exchanger equations developed in the last section apply only to cases where the steam is saturated at all locations in the condenser. If superheated steam was lead directly into the condenser, as shown in the left part of fig. 3-17, the heat trans-

fer in the first part of the condenser would take place with a poorer heat transfer coefficient on the steam side (forced convection instead of steam condensation). On the other hand in this case the average temperature difference between the steam and water sides is larger than at the point where the condensation begins. The net effect of superheated steam on the overall heat transfer could be both positive or negative. Another solution, when superheated steam is led to the condenser, though not commonly adopted, is to recycle part of the condensate to a spray cooler (the right part of fig. 3-17), where the steam is desuperheated. In this way it can be ensured that the steam flow entering the condenser is saturated, so that the heat exchanger equations developed in the last section will apply.

The two enthalpies plotted in fig. 3-16 change with condensing temperature in a way which is quite close to linear relationships, as long as the turbine efficiency is considered constant. This means that without great error we may linearize the influence of changes in condensing temperature on the steam cycle to simple analytical expressions.

Adopting this linearization we may formulate the following equations:

$$P_e^* = P_{e,o}^* (1 + a (t_c - t_{c,o})) m_s^* \quad (3-115)$$

$$P_{q1}^* = P_{q1,o}^* (1 + b (t_c - t_{c,o})) m_s^* \quad (3-116)$$

$$P_{eb}^* = P_{eb,o}^* (1 + e (t_c - t_{c,o})) m_s^* \quad (3-117)$$

Here the asterix * is added to symbols for energy flows to indicate that they have been made dimensionless by dividing them with the reference value, $P_{q2,o}$, of the heat requirement of the DH buildings. Note that it is not the heat flow, P_{q1} , added from the generation plant to the DH network that is used in the division. The steam flow, m_s , has been made dimensionless by dividing it with its value, $m_{s,o}$, at reference load conditions.

P_e is the output generated by the turbine. When the power requirement P_{eb} of the feedwater pump is subtracted, we get the net output from the steam cycle:

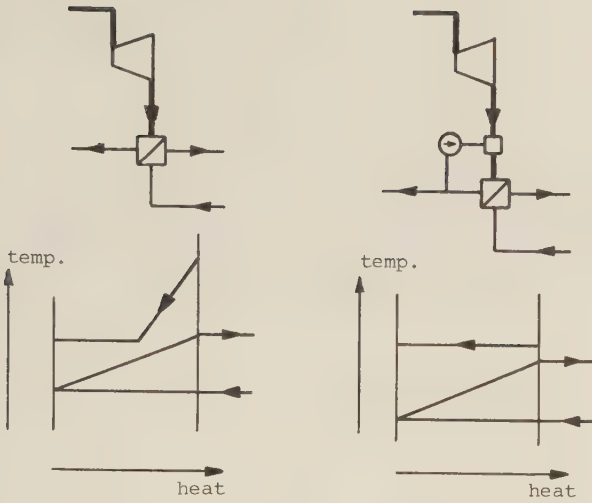


Fig. 3-17

Alternative ways of handling superheated steam from turbine exhaust.

Superheated steam led directly into condenser

Desuperheating of steam by condensate injection

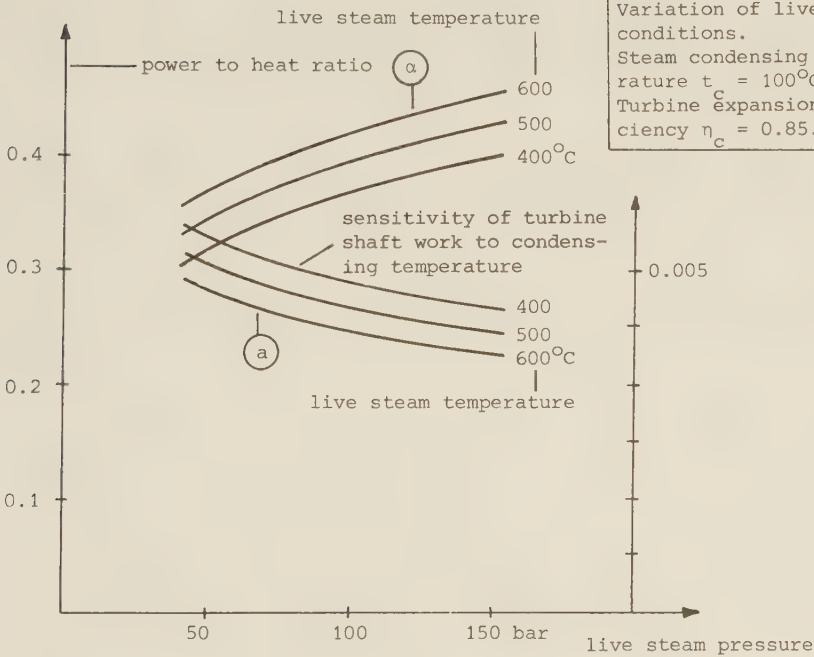


Fig. 3-18

Variation of live steam conditions.
Steam condensing temperature $t_c = 100^\circ\text{C}$.
Turbine expansion efficiency $\eta_c = 0.85$.

$$P_{e1}^* = P_e^* - P_{eb}^* \quad (3-118)$$

(powers again made dimensionless by dividing them by $P_{q2,o}$).

Two of the influence coefficients, a and e , are always negative, while b may be either positive or negative as can be read from fig. 3-16. Here the lines for h_1 and h_4 converge with increasing t_c for a turbine efficiency of 0.75, and diverge when the efficiency is 0.95. In general, b is found to vary round 0.

By dividing P_e by P_{q1} we get the power-to-heat ratio:

$$\alpha = P_e / P_{q1} = \alpha_o \frac{1 + a (t_c - t_{c,o})}{1 + b (t_c - t_{c,o})} \quad (3-119)$$

Except for theoretical cases of large negative values of b , the ratio is seen to decrease at higher condensing temperature t_c . When b is positive, the amount of heat delivered by each unit of steam to the condenser increases with condensing temperature. Thus, for constant P_{q1} , P_e is reduced, not only because of a higher back-pressure, but also by a reduced steam flow through the turbine.

Fig. 3-18 shows how α and a vary, when for constant t_c and constant turbine efficiency both the admission pressure and temperature take different values. The lower the admission conditions are, the lower is the power-to-heat ratio, and the greater is the sensitivity to changes in condensing temperature. α and a are seen to change almost inversely, which is explained by writing:

$$a = \frac{dh_4/dt_c}{h_3 - h_4} \quad (3-120)$$

and:

$$\alpha = \frac{h_3 - h_4}{h_4 - h_1} \quad (3-121)$$

which may be combined to:

$$a = - \frac{dh_4/dt_c}{h_4 - h_1} \frac{1}{\alpha} \quad (3-122)$$

In the h - s Mollier diagram the straight-line isothermals converge at low entropies, which means that the ratio, by which $1/\alpha$ is multiplied in the above equation, is roughly a constant. Thus, a and α will vary approximately inversely.

In fig. 3-18 the values for a are around $1/200^\circ\text{C}$, the inverse of which is smaller than the temperature difference of 400°C between steam admission temperature and reference condensing temperature of 100°C . The explanation is that the limiting case of vanishing turbine shaft work corresponds to a condensing temperature of not 500°C , but 310°C , which is the boiling temperature of water at 100 bar, the admission pressure in the numerical example here.

In fig. 3-16 the variation of h_4 with t_c is seen to be approximately linear when the efficiency of the turbine expansion is constant. For real turbines the efficiency can be expected to vary. Fig. 3-19 shows an estimate of possible variations in the turbine condition curve in the case of nozzle governing. As in fig. 3-15 only static condition points are indicated. For a true picture of the efficiency, the effects of variation in end-velocity losses should be included, as pointed out at the beginning of this section.

Accurate and explicit accounts on back-pressure turbine efficiency, varying with independent changes in steam flowrate and in condensing temperature, do not seem to have been reported in literature, although the theoretical tools necessary for such an investigation are available from classical turbine theory [3-15]. Cizmar [3-16] utilized this theory to investigate variations in both turbine efficiency and in plant efficiency of a condensing turbine equipped with various types of turbine control. Kroon and Tobiasz [3-17] mapped the performance of an assembly of uniform turbine stages under varying load conditions. Their calculations did not include a control stage, so that they pertain to only part of the expansion line covered by fig. 3-19. Recently a manufacturer [3-18] in a diagram gave some examples of realistic

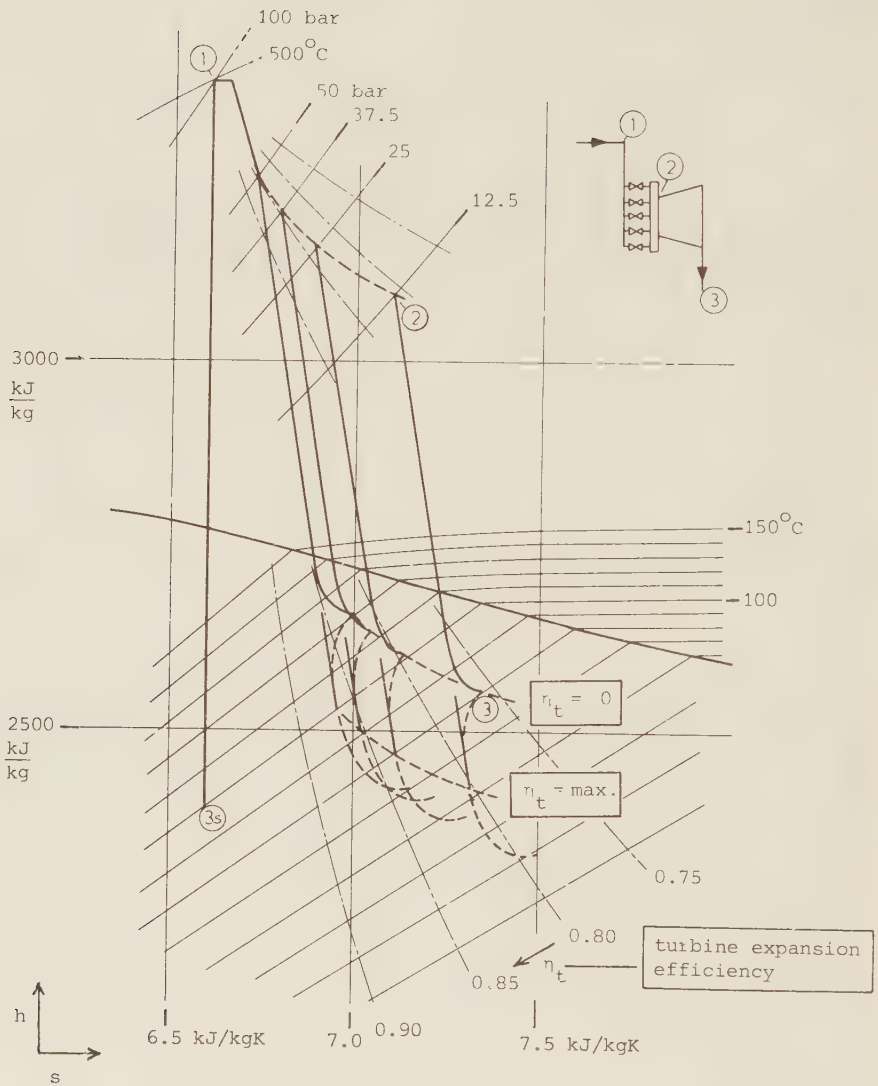


Fig. 3-19

Turbine condition curves in Mollier diagram. Expansion end points indicated for various steam flowrates and for various back-pressures.

expansion lines for a back-pressure turbine. In their general character they coincide with those given in fig. 3-19.

From fig. 3-19 consequences of combined variations of turbine back-pressure and of steam flow can be read. For each steam flow, m_s^* , a dotted line below the saturation line indicates expansion end-points for different condensation temperatures, given by the isothermals in the diagram. Also indicated as dotted lines are points representing constant isentropic efficiencies of the total expansion from the admission condition (1 with a circle around the figure). Position 2 at intermediate pressure represents the steam condition after the control stage. As can be seen, this point moves to lower pressures and higher entropies at part loads. Also, dotted lines connect points of constant efficiency for the control stage alone. In the numerical example of the figure, all stages are designed to operate at maximum efficiency at the reference load, here defined as the case of $m_s^* = 1$ (maximum steam flow) and $t_c = 100^\circ\text{C}$. The efficiency of the control stage is significantly lower than that of all other stages at reference load. The expansion from 2 to 3 is a straight line. In practice, somewhat lower efficiencies will be seen for stages below the saturation line, due to differences in kinematic trajectories of saturated steam and condensed droplets.

From fig. 3-19 the following results can be read: When m_s^* is kept constant = 1 (full load), and the condensing temperature t_c is increased from its reference value, there is initially almost no deterioration of the efficiencies of the turbine stages, the effect on the condition curve is only that the straight line of expansion is shortened. Nevertheless, the efficiency of the total expansion decreases a little, since a relatively larger part of the total expansion takes place in the control stage with its lower efficiency. When the increase in t_c becomes greater, the line connecting end-points bends outwardly, and so does the condition curve itself. This means that when increased irreversibilities are induced in the steam expanding in the turbine, they are concentrated to the lower end of the expansion, and in particular the very last stage is affected. At some point of raised back-pressure (in the figure where $t_c = \text{approx. } 123^\circ\text{C}$) the end-point tangent becomes horizontal, i.e. the stage work of the last stage vanishes, it acts as no more than a throttle on the steam flow. Possible effects of a further rise in condensing temperature have not been shown in

the figure; it is possible that the last stages may act as compressor stages of low efficiency at high condensing temperatures. When the condensing temperature is lowered instead of increased from its reference value, the expansion end-point, a reversed pattern is found. I.e., at moderate temperature changes a prolonged expansion takes place in the turbine with nearly unaffected last-stage efficiencies, and at greater changes in condensing temperature a progressive deterioration of operation of the last stage is found, until finally the stage work of the last stage vanishes.

Turning now to cases of m_s^* being smaller than 1 (part loads), we may observe initially that the pressure after the control stage is unaffected by the condensing temperature. This is a consequence of the so called 'ellipse law' according to Stodola. In the special case of a back-pressure much lower than the admission pressure, this law states that the steam flow through a stage assembly (such as the expansion from point 2 to point 3 in fig. 3-19) varies proportionally to the admission pressure. When the condensing temperature t_c is varied at part load, the expansion end-point follows a curve similar to that which is found at full load. However, at part load the point of maximum last stage efficiency shifts to lower condensing temperatures. The reason for this shift is found in turbine stage theory, see e.g. Traupel [3-15]. Here, the stage efficiency is shown to vary with the dimensionless steam flow:

$$\phi = \frac{m_s v_2}{u_2 \varepsilon \Omega_2} \quad (3-123)$$

where v_2 is the specific steam volume at the stage exit, u_2 is the peripheral speed at the exit, ε is the part of the flow perimeter loaded by the steam, and Ω_2 is the full flow area at stage exit. Thus, if the stage efficiency is supposed to remain constant for a turbine rotating at constant speed, the product $m_s v_2$ must also remain constant. In fig. 3-19 the dotted line for $\eta_t = \max$, connects points for which the last stage operates at maximum efficiency.

It can be seen that in all cases where the efficiency of the turbine expansion after the control stage deteriorates, as a result of altered operational conditions, the increased irreversibilities are concentrated to the low-

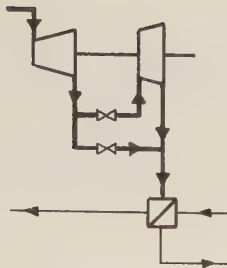


Fig. 3-20 Back-pressure turbine with low-pressure by-pass option.

pressure part of the expansion. The reason is that changes in steam pressure are located to the last stages, as is seen from Stodola's 'ellipse law', applied to both the entire stage assembly after the control stage, and to the low-pressure part of the expansion [3-15].

The fact that a combination of decrease in steam flow and a lowering of the condensing temperature may be favourable in terms of turbine efficiency, may be of considerable practical interest. It is customary and advantageous, as model calculations in later chapters will show, to operate DH systems with a supply temperature which is increased with the heat loading of the system. Qualitatively, this fits in well with the kind variation in turbine efficiency indicated by fig. 3-19. Of course this does not mean that in all cases the matching will be perfect.

The variations in turbine expansion lines shown in fig. 3-19 have here been shown for one special case of turbine design. With other values for design parameters it is possible to change the variation in turbine efficiency. For instance, the control stage may be designed to have its maximum stage efficiency at some part load, not at full load as in the figure. This could reduce the range of variation for the efficiency of the entire turbine.

Another possibility of reducing variations in turbine efficiency by design measures, would be to equip the turbine with one or more low-pressure by-passes, see fig. 3-20. When, for instance, at high back-pressures, the number of stages comprising the full turbine is larger than optimal, the low-pressure part of the turbine may here be by-passed. In this way low efficiency performance of the last stages may be avoided within a large range of operating conditions.

Concluding the discussion we may state that significant variations in turbine efficiency are quite possible, that a detailed calculation of their magnitude is extremely complicated, but that at the design stage several possibilities exist to manipulate the variation in efficiency.

3.5 The Steam Plant Boiler

Since in this work we have chosen to focus our investigations of system performance to changes in net electric output, expressed as the difference between steam cycle output and pumping power for the DH system, no detailed account of boiler performance is needed. However, since in section 4.1 consequences for fuel heat rate of variations in DH control strategy are considered to some extent, a rough estimate of changes in boiler efficiency will serve a useful purpose.

The energy balance for the boiler may be written as:

$$m_f H + P_{ea} = P_q + P_{qa} \quad (3-124)$$

where: $m_f H$ = fuel heat rate with H as higher or lower heating value of fuel, depending on which one is appropriate,

P_{ea} = auxiliary power needed for mechanical devices (fans, fuel preparation, etc),

P_q = heat rate added to steam cycle,

P_{qa} = rate of boiler energy losses.

Normally the boiler energy losses, usually amounting to around 10% for a large boiler, will be much larger than the auxiliary mechanical power, being in the order of 1% of the heat rate. In special circumstances, such as relatively low prices for electricity, a reduction of P_{qa} could be considered at the cost of a higher value for P_{ea} by increasing gas velocities over heat transfer surfaces.

But normally the question of how close variations in P_q are reproduced as variations in $m_f H$ will be a matter of examining variations in P_{qa} . P_{qa} may be broken down into three components:

$$P_{qa} = P_{qunburnt} + P_{qgrad} + P_{qstack} \quad (3-125)$$

Of these terms the radiation losses P_{grad} through boiler walls, and the stack loss, P_{qstack} , will normally dominate over the loss P_{qunburnt} due to unburnt fuel. Since P_{grad} is related to the surface/volume ratio of the boiler, it will generally decrease in magnitude with the size of the boiler. In the case of a big power station boiler, P_{grad} and P_{stack} may be of the same order of magnitude. Whereas P_{grad} for a given boiler is relatively independent of load, the size of P_{qstack} will vary with operation conditions.

Neglecting variations in combustion gas specific heat c_{pa} , the stack loss may be written as:

$$P_{\text{qstack}} = m_a c_{\text{pa}} (t_{\text{stack}} - t_a) \quad (3-126)$$

Fig. 3-21 shows a typical temperature distribution in the low-temperature section of a boiler with an economiser and an air preheater. The incoming water temperature, t_c , to the economiser has been assigned a low value, corresponding to a steam power cycle without regenerative feedwater heating. The figure indicates two types of measures for control of the stack temperature: Economiser by-pass and recirculation of preheated air.

When no such control measures are applied, the stack temperature becomes a determined function of the load conditions of the boiler:

$$t_{\text{stack}} = t_{\text{stack}}(m_s^*, t_a, t_c) \quad (3-127)$$

i.e. the dimensionless steam flowrate m_s^* , characterizing the size of the boiler load, the ambient temperature t_a of the incoming air, and the temperature t_c of the incoming feedwater. Fig. 3-22 represents an estimate of typical sensitivities of t_{stack} to changes in the three parameters. A lowering of m_s^* lowers t_{stack} , due to a more favourable heat transfer surface/mass flowrate ratio for all heat transfer surfaces of the boiler, lowering all combustion gas temperatures throughout the boiler. t_{stack} is seen to follow changes in t_a to some degree; however, with a lowering of t_a , the temperature difference $t_{\text{stack}} - t_a$, determining the stack loss, will increase. Finally t_{stack} is seen to display only a small sensitivity to changes in t_c . Had t_c been higher than the temperature t_{air} of the preheated air,

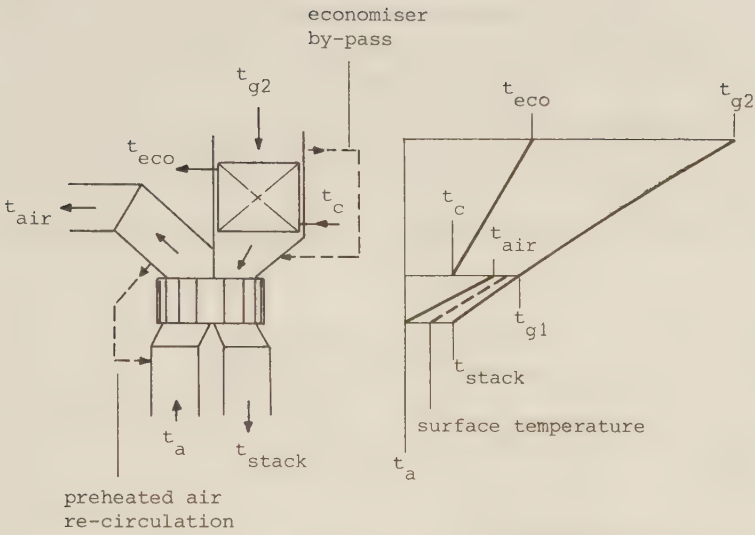


Fig. 3-21 Schematic of low-temperature section of steam plant boiler.

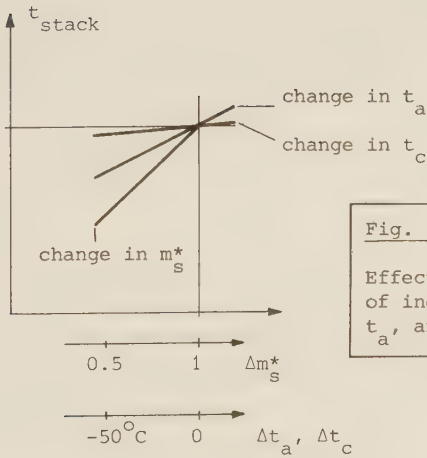


Fig. 3-22

Effects on stack temperature of independent changes in m_s^* , t_a , and t_c .

such as is typically the case with a steam cycle with regenerative feedwater heating, then the sensitivity would be greater. With no feedwater heating present, t_c is equal to the condensing temperature of the steam in the plant condenser. Therefore, at changes in DH network operation, variations in the incoming water temperature to the boiler will not be far from variations in the supply water temperature t_{out1} . With feedwater heaters present, variations in condensing temperature will be damped in the successive heating stages in the feedheaters. Instead, the incoming water temperature to the boiler becomes a function of the load: At lower load the bled steam flows to the feedheaters go down, so that the final feedwater temperature becomes lower.

Normally the stack temperature is not kept floating freely. Usually very low values will be unacceptable because of the risk of low-temperature corrosion. This is likely to occur on cold surfaces in contact with combustion gases, particularly if the surface temperature is below the dew point of the water present in the gas, i.e. in the order of $30 - 50^{\circ}\text{C}$. In fig. 3-21 the mean temperature between combustion gas and air for preheating, varying throughout the preheater, has been indicated. When the heat transfer rates of the two sides of the preheater are equal, the mean temperatures may be considered representative of the surface temperature. In the case of a preheater of the rotary Ljungström type the actual surface temperature pulsates, so that an exact definition of a critical surface temperature becomes difficult. It can be seen that the lowest surface temperature occurs at the combustion gas exit of the preheater, so that as a first criterion for avoidance of low-temperature corrosion we may write:

$$t_{\text{mean}} = \frac{t_{\text{stack}} + t_a}{2} > \text{water dew point} \quad (3-128)$$

If the boiler is operated without any stack temperature control, the full load value of t_{stack} must be selected so high that at unfavourable low load conditions the critical surface temperatures does not fall below the dew point. In this respect it is favourable that for a CHP plant low loads (i.e. small values of m_g^*) occur at high values of t_a , allowing for a relatively low t_{stack} , according to eq. (3-128).

Nevertheless a free-floating stack temperature will normally cause the full load value of the stack temperature to be selected with a safety margin, giving an increase of the stack loss at high loads. Therefore there is the motivation to equip the boiler with control devices, as indicated in fig. 3-21, so that the surface temperature t_{mean} is kept constant. For example, at lower loads the by-pass flow passing the economiser can be increased, counteracting the temperature lowering following the improved heat transfer at smaller flowrates by mixing with by-pass gas.

Approximating $P_{\text{ea}} = P_{\text{gunburnt}} = 0$, we may write for the boiler efficiency:

$$\eta_b = 1 - \frac{P_{\text{grad}}}{m_f H} - \frac{m_a c_{\text{pa}}}{m_f H} (t_{\text{stack}} - t_a) \quad (3-129)$$

From this expression we can make the following predictions for the variation of boiler efficiency in the case of a CHP plant and of a boiler with stack temperature control:

- If the DH supply temperature is varied at constant heat load the boiler efficiency will be virtually unaffected.
- If the load is decreased then the size of the second of the three terms in eq. (3-129) will increase. This will to some extent be counteracted by the diminishing of the temperature difference in the third term.

On the whole, variations in boiler efficiency with load conditions can be expected to be small.

In cases where the auxiliary power P_{ea} is significant, it will be of interest to discuss how variations in P_{ea} affect $m_f H$. In a first approximation we may assume that P_{ea} varies with the boiler load according to:

$$P_{\text{ea}}^* = P_{\text{ea},0}^* m_s^3 \quad (3-130)$$

Part of P_{ea} may be regarded as recovered by the boiler. Thus, power added to coal for crushing it in mills may be considered to increase the heating value of the fuel. Also, power needed for forced draught fans may be said

to be recovered, since the temperature rise at the compression does not increase the stack temperature with consequent increase in the stack loss. In fact more than 100% of power for forced draught fans may be considered recovered, since the rise in incoming air temperature allows for a corresponding lowering of the stack temperature to give an unchanged lowest surface temperature in the air heater. Conversely, power needed for induced draught fans may be considered lost, since in that case the temperature rise at the compression leads to a higher gas temperature at the entrance to the chimney, for a given preheater surface temperature. Thus, the question of to what extent P_{ea} is recovered in the system is not so simple as with the pumping power P_{ec} for the DH system, where the medium circulates in a closed loop, and not in an open loop like the combustion gas.

4. PERFORMANCE OF SIMPLE SYSTEM MODEL

4.1 Load Performance of Simple System Model

In Chapter 3 equations were developed for the load performance of the various system elements. In the present section we shall investigate the load performance of our simplest system model, which was presented in Chapter 2, i.e. a system with no supplementary heat-only boiler, and with no hot consumption water supply for the building.

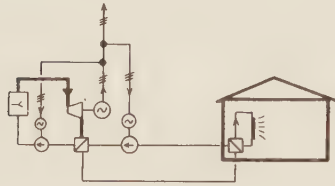
Table 4-1 summarizes the performance equations for the elements of the system, developed previously, with variables in dimensionless form, and with values of parameters related to a reference state of the system, indicated by the index o. Table 4-2 summarizes the mathematical idealizations adopted in table 4-1. The ambient temperature t_a is the only exogenous system variable. The system has a single degree of freedom, allowing for different operation modes of the DH system at a given heat load. This degree of freedom may be characterized, either by the DH mass flowrate, m^* , or by the supply temperature (t_{out1} or t_{out2}). To begin with we shall select m^* , since it will allow us to calculate P_{e2} straightforwardly, without iterations. System characteristics with the supply temperature characterizing the single degree of freedom will be shown later.

Fig. 4-1 shows the result of a calculation based on the system of equations in table 4-1, with parameter values given by table 4-3. Depicted are five important variables, the net electric output, the DH pumping power, the heat loss from the DH system, and the supply and return temperatures of the DH system (as measured at the consumer connection).

The supply and return temperatures are seen to vary in accordance with the performance shown in section 3.3. The pumping power varies with the cube of the mass flowrate and is independent of the heat load. The heat loss goes down with increasing mass flowrate at a given heat load, since the supply temperature goes down. The fact that the total DH heat loss is the sum of losses from supply and return pipes suggests that in cases with a much poorer insulation of the return pipe than of the supply, the total heat loss could theoretically increase with a greater mass flowrate (increasing the return temperature). The internal heat flow between supply and return pipes, not shown in the figure, diminishes with increasing flow-

Table 4-1

System of equations describing different operational modes for simple model of DH system with back-pressure turbine.



Dimensionless variables:

$$P_{q2}^* = P_{q2}/P_{q2,o}$$

$$P_{q1}^* = P_{q1}/P_{q2,o}$$

...

$$P_{e2}^* = P_{e2}/P_{q2,o}$$

$$m^* = m/m_o$$

$$m_s^* = m_s/m_{s,o}$$

$$U_1^* = \frac{U_1 A_1}{mc}$$

$$U_2^* = \frac{U_2 A_2}{mc}$$

$$P_{q2}^* = 1 + i(t_a - t_{a,o}) \quad (3-1)$$

$$t_1 = t_h + (t_{h,o} - t_h) P_{q2}^{*1/n} \quad (3-9)$$

$$P_{ec}^* = P_{ec,o}^* m^{*3} \quad (3-11)$$

$$U_2^* = \frac{U_{2,o}^*}{m^*} \frac{1 + x_2}{m^* - P_2 + x_2} \quad (3-102)$$

$$t_{out2} = t_1 + (t_{out2,o} - t_{1,o}) \frac{P_{q2}^* - zP_{ec}^*}{1 - zP_{ec,o}^*} \frac{1 - \exp(-U_{2,o}^*)}{1 - \exp(-U_2^*)} \quad (3-100), R=1$$

$$t_{in2} = t_{out2} - (t_{out2} - t_1) (1 - \exp(-U_2^*)) \quad (3-103), R=1$$

$$t_{in2,o} = t_{out2,o} - (t_{out2,o} - t_{1,o}) (1 - \exp(-U_{2,o}^*))$$

$$j = P_{qIO,o}^* / (t_{out2,o} - t_s) \quad (3-27)$$

$$P_{qIO}^* = P_{qIO,o}^* + j(t_{out2} - t_{out2,o}) \quad (3-28)$$

$$k = P_{qIIIO,o}^* / (t_{in2,o} - t_s) \quad (3-29)$$

$$P_{qIIIO}^* = P_{qIIIO,o}^* + k(t_{in2} - t_{in2,o}) \quad (3-30)$$

$$l_{10} = P_{qIIIO,o}^* / (t_{out2,o} - t_{in2,o}) \quad (3-31)$$

(to be continued)

Table 4-1, continued

$$l_{01} = -l_{10} \quad (3-32)$$

$$P_{qIII}^* = P_{qIII,o}^* + l_{10}(t_{out2} - t_{out2,o}) + l_{01}(t_{in2} - t_{in2,o}) \quad (3-33)$$

$$P_{qc}^* = P_{qIO}^* + P_{qIIO}^* \quad (3-34)$$

$$P_{qc,o}^* = P_{qIO,o}^* + P_{qIIO,o}^* \quad (3-35)$$

$$P_{q1}^* = P_{q2}^* + P_{qc}^* - P_{ec}^* \quad (3-36)$$

$$P_{q1,o}^* = 1 + P_{qc,o}^* - P_{ec,o}^* \quad (3-37)$$

$$w = 1 - x - y - z \quad (3-12)$$

$$t_{in1} = t_{in2} + (t_{out2} - t_{in2}) \frac{yP_{ec}^* - P_{qIIO}^* + P_{qIII}^*}{P_{q2}^* - zP_{ec}^*} \quad (3-38)$$

$$t_{in1,o} = t_{in2,o} + (t_{out2,o} - t_{in2,o}) \frac{yP_{ec,o}^* - P_{qIIO,o}^* + P_{qIII,o}^*}{1 - zP_{ec,o}^*} \quad (3-39)$$

$$t_{out1} = t_{in1} + (t_{out2} - t_{in2}) \frac{P_{q1}^* + wP_{ec}^*}{P_{q2}^* - zP_{ec}^*} \quad (3-40)$$

$$t_{out1,o} = t_{in1,o} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1,o}^* + wP_{ec,o}^*}{1 - zP_{ec,o}^*} \quad (3-41)$$

$$U_1^* = \frac{U_{1,o}^*}{m^*} \frac{1 + x_1}{-P_1^* + x_1}$$

$$t_c = t_{in1} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1}^* + wP_{ec}^*}{m^*} \frac{1/(1 - zP_{ec,o}^*)}{1 - \exp(-U_1^*)} \quad (3-114)$$

$$t_{c,o} = t_{in1,o} + (t_{out2,o} - t_{in2,o}) (P_{q1,o}^* + wP_{ec,o}^*) \frac{1/(1 - zP_{ec,o}^*)}{1 - \exp(-U_{1,o}^*)}$$

$$m_s^* = P_{q1}^* / (P_{q1,o}^* (1 + b(t_c - t_{c,o}))) \quad (3-116)$$

$$P_e^* = P_{e,o}^* m_s^* (1 + a(t_c - t_{c,o})) \quad (3-115)$$

$$P_{eb}^* = P_{eb,o}^* m_s^* (1 + e(t_c - t_{c,o})) \quad (3-117)$$

$$P_{e1}^* = P_e^* - P_{eb}^* \quad (3-118)$$

$$P_{e2}^* = P_{e1}^* - P_{ec}^*$$

Table 4-2

Summary of mathematical idealizations
made in table 4-1.

- * Building heat load and heat loss components of DH system assumed to vary linearly with temperature differences.
- * DH heat loss components calculated from supply and return temperatures at consumer's connection, not from varying water temperatures along pipes.
- * DH water flow in pipelines and in heat exchangers assumed to be completely turbulent.
- * Heat transfer coefficients in heat DH exchangers assumed to be unaffected by variations in water temperature level.
- * Heat generated by DH water pressure drops in heat exchangers assumed to be transferred without heat flow resistance in the DH water.
- * Pressure rise in DH pump assumed to take place without irreversibilities.
- * Steam condensing temperature influence on steam cycle linearized.
- * Turbine efficiency assumed to be unaffected by variations in steam flowrate.
- * Effects of variation in turbine steam end-velocity losses and of steam-side condenser pressure drop neglected.
- * Conversion losses between thermodynamical shaft work, mechanical shaft work, and electrical power, as well as losses of transformation and transmission of electrical power, all neglected.

Fig. 4-1

Simple system model load performance

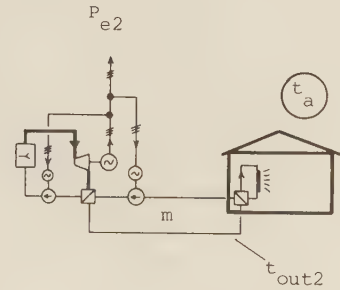
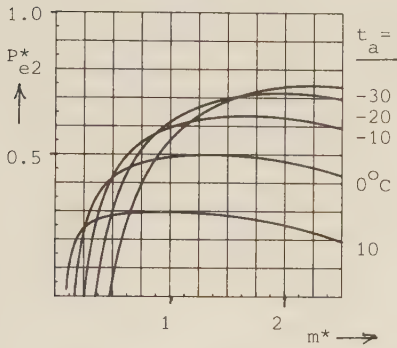
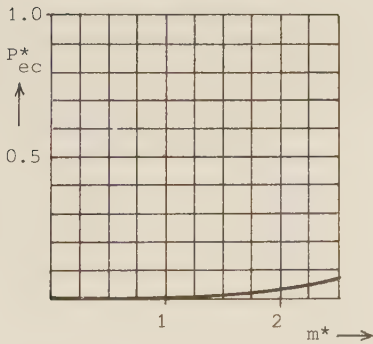
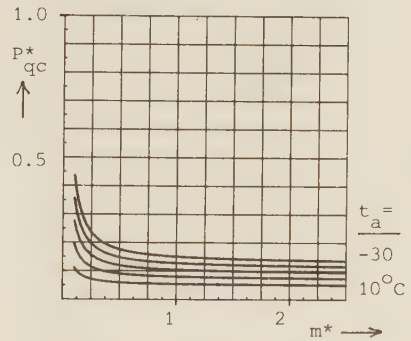
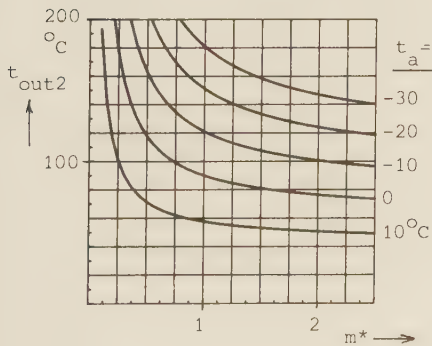
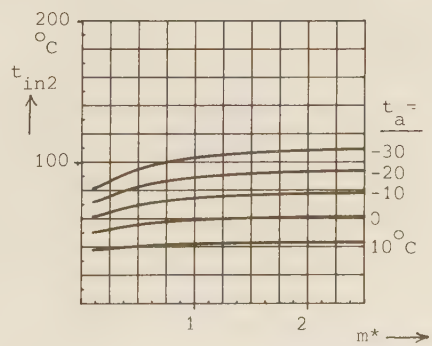
Abscissa: DH mass flowrate m^* Curve parameter: ambient temperature t_a Net electric output:DH pumping power:DH heat loss:Supply water temperature:Return water temperature:

Table 4-3

Input values for parameters in table 4-1, valid for all results shown in diagrams in section 4.1.

Constants:	Values defining reference conditions:
$t_s = 0^{\circ}\text{C}$	$t_{a,o} = 0^{\circ}\text{C}$
$t_h = 20^{\circ}\text{C}$	$t_{1,o} = 50^{\circ}\text{C}$
$n = 1.3$	$t_{\text{out}2,o} = 90^{\circ}\text{C}$
$x_1 = 0.1$	$U_{1,o}^* = 1.5$
$x_2 = 1.0$	$U_{2,o}^* = 1.5$
$p_1 = 0.5$	$P_{q\text{IO},o}^* = 0.04$
$p_2 = 0.5$	$P_{q\text{IIO},o}^* = 0.04$
$x = 0.25$	$P_{q\text{III},o}^* = 0.04$
$y = 0.25$	$P_{ec,o}^* = 0.005$
$z = 0.25$	$P_{eb,o}^* = 0.005$
$a = -0.005$	$P_{e,o}^* = 0.5$
$b = 0$	
$e = -0.001$	
$i = -0.05$	

rate, as the difference between supply and return temperatures becomes smaller. The effect of increased heat load (lower t_a) on the total heat loss is seen to be an increase, due to higher supply and return temperatures.

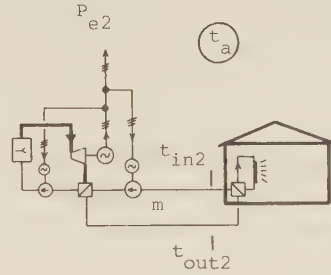
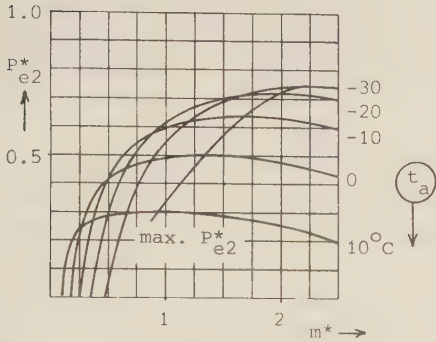
It is seen that for each ambient temperature, t_a , the net electric output P_{e2}^* has a maximum. This maximum is the result of two counteracting effects: With increasing mass flowrate m^* the pumping power P_{ec}^* increases, but at the same time the supply temperature is lowered, so that the enthalpy drop over turbine is increased, giving a greater electric output P_e^* from the turbine.

Fig. 4-2

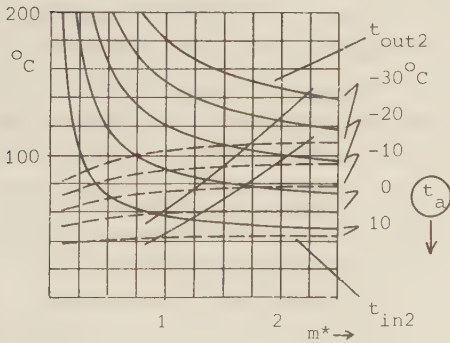
Load performance of simple system model.

DH operational modes to maximize net electric output P_{e2}^* at varying heat load.

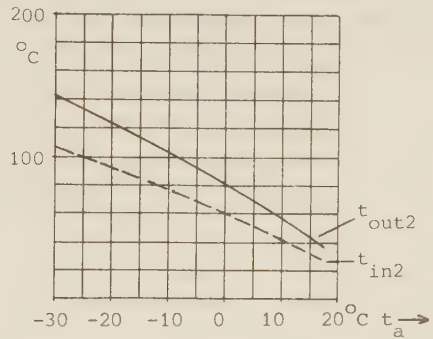
Net electric output:



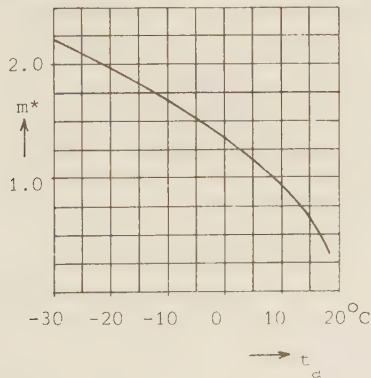
Water temperatures:



Water temperatures when $P_{e2}^* = \text{max}$:



Water flowrate when $P_{e2}^* = \text{max}$:



From the figure it can also be seen that the value of m^* maximizing P_{e2}^* increases with larger heat loads. At the same time the maximum values of P_{e2}^* themselves also increase, though the marginal gain diminishes progressively as the heat load goes up. Extending this tendency beyond the greatest heat load in the figure, i.e. to ambient temperatures below -30°C , would give a global maximum for P_{e2}^* as a function of both t_a and m^* . At still higher loads P_{e2}^* would decrease, even if at each heat load m^* is selected to maximize P_{e2}^* . In a less favourable numerical example, for instance in the case of a DH system with a smaller pipe diameter, the heat load for global maximum of P_{e2}^* may be found at higher ambient temperature.

In the diagram at the top of fig. 4-2 the curve for maximum P_{e2}^* at different heat loads has been plotted. The two diagrams below indicate corresponding variations of supply and return temperatures of the DH system. The diagram at the bottom of the figure shows how m^* must change with t_a to maximize P_{e2}^* . From the figure it can be concluded that with an increasing heat load, both the DH mass flowrate and the supply temperature should be raised. At the same time the return temperature also goes up, though less rapidly than the supply temperature, so that there is an increase in the temperature difference with increasing heat load. Had the mass flowrate been kept constant, sometimes referred to as pure qualitative DH network control (in contrast to pure quantitative type of control with constant supply temperature), then the supply temperature would have to rise more rapidly with the heat load.

It is justified to ask if maximum P_{e2}^* is the most appropriate criterion when selecting values for m^* and t_{out2} . Since at variations in DH mass flowrate, in addition to P_{e2}^* , the heat rate $m_f H$ of the plant varies, this may also be taken into account. The heat input P_q to the steam cycle is related to $m_f H$ by the boiler efficiency η_b by the relation:

$$P_q^* = \eta_b (m_f H)^* \quad (4-1)$$

For P_q we may also write:

$$P_q^* = P_{q2}^* + P_{e2}^* + P_{qc}^* \quad (4-2)$$

When m^* is varied at constant P_{q2}^* we have for the variations in P_q^* :

$$\Delta P_q^* = \Delta P_{e2}^* + \Delta P_{qc}^* \quad (4-3)$$

Assuming that the efficiency η_b of the boiler is constant at variations in m^* (cf. section 3.5), a specific economic value, c_f , may be assigned to P_q^* . Similarly, a specific economic value c_e may be assigned to P_{e2}^* . Therefore economic consequences of variations in m^* may be expressed as the difference:

$$\Delta C = c_f \Delta P_q^* - c_e \Delta P_{e2}^* \quad (4-4)$$

which by inserting eq. (4-2) gives;

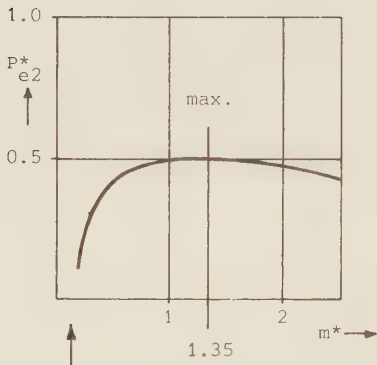
$$\Delta C = c_f (\Delta P_{qc}^* - (\frac{c_e}{c_f} - 1) \Delta P_{e2}^*) \quad (4-5)$$

Normally electric energy is rated higher than heat, so that in the last equation the coefficient in front of ΔP_{e2}^* will be positive.

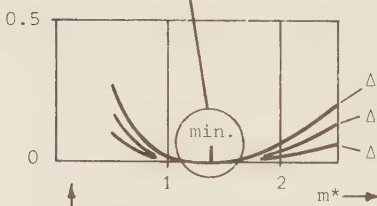
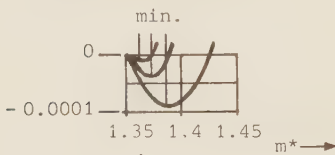
In the diagram at the bottom left of fig. 4-3 is shown how the cost ratio $\Delta C/c_f$ varies for different values of the coefficient for ΔP_{e2}^* , the value of m^* maximizing P_{e2}^* as a reference in the special heat load case of $t_a = 0$. Economic optima are found where $\Delta C/c_f$ curves are at minimum.

From the figure it can be concluded that taking variations in heat rate of the plant into consideration when deciding on m^* , leads to somewhat higher values of m^* than those maximizing P_{e2}^* . However, the increase in m^* will be small, so long as the specific cost c_e for electricity is at least twice the specific cost c_f for heat (supplied by the fuel). Also, the improvement in total cost by increasing m^* beyond the value maximizing P_{e2}^* is only slight. Therefore, when in the following sections we neglect variations in the economic value of the heat input by the fuel, this will not be a serious simplification.

At the bottom of fig. 4-3 is shown, for comparison with the case of our back-pressure plant, the result of economic optimizations of DH mass flow-rates in the case of a heat-only boiler as the heat source for the DH sys-

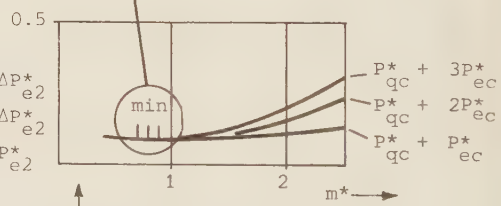
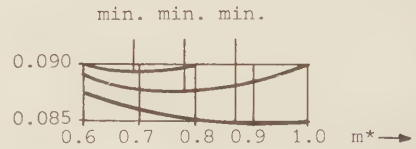
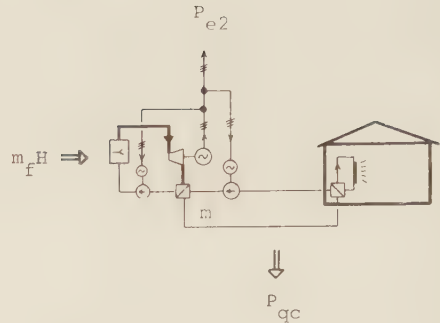


Back-pressure plant (max. P_{e2}^*)



Back-pressure plant

(Variations in net electric output weighed against variations in plant heat rate)



Heat-only boiler

(Variations in heat rate weighed against variations in DH pumping power)

Fig. 4-3 Optimum choice of DH water flowrate according to various criteria.

$$t_a = 0^\circ\text{C}.$$

tem. Instead of eqs. (4-1) and (4-2) we then have:

$$P_{q1}^* = \eta_b (m_f H)^* \quad (4-6)$$

and:

$$P_{q1}^* = P_{q2}^* - P_{ec}^* + P_{qc}^* \quad (4-7)$$

As a relevant cost for optimization we may put:

$$\Delta C = c_f \Delta P_{q1}^* + c_e \Delta P_{ec}^* \quad (4-8)$$

(again neglecting variations in boiler efficiency). Inserting eq. (4-7) gives:

$$\Delta C = c_f (\Delta P_{qc}^* + (\frac{c_e}{c_f} - 1) \Delta P_{ec}^*) \quad (4-9)$$

The diagram at the bottom right of fig. 4-3 shows that the case of a heat-only boiler leads to lower economically optimal mass flowrates than with a back-pressure plant. For both types of heat generation, a lowering of the supply temperature at a higher flowrate gives a smaller heat loss and a higher pumping loss. In the case of the back-pressure plant we have the additional effect of a higher electric output, i.e. a further incentive to lower the supply temperature which explains the difference in optimal mass flowrate.

Introducing some simplifications we may arrive fairly easily at an analytical expression for the DH mass flowrate of the maximum net electric output of the back-pressure plant. Since at maximum P_{e2}^* the heat loss P_{qc}^* varies only a little, we may approximate:

$$P_{qc}^* = 0 \quad (4-10)$$

Observing that the supply temperature normally varies only a little from the generation plant to the consumer, we may make the simplification:

$$t_{out} = t_{out1} = t_{out2} \quad (4-11)$$

The return temperature is seen to vary very moderately, except at low mass flowrates, so that it will be no great error to put:

$$t_{in} = t_{in1} = t_{in2} = \text{constant} \quad (4-12)$$

For the steam cycle the influence coefficient b , expressing variations in heat flow P_{q1}^* with condensing temperature, is small, so that we may put:

$$b = 0 \quad (4-13)$$

So long as the difference between steam condensing temperature and DH supply temperature is small, variations in condensing temperature will closely follow variations in supply temperature, so that we may write approximately:

$$\frac{dt_c}{dm^*} = \frac{dt_{out}}{dm^*} \quad (4-14)$$

At maximum P_{e2}^* we have:

$$\frac{dP_{e2}^*}{dm^*} = \frac{dP_e^*}{dm^*} - \frac{dP_{ec}^*}{dm^*} = 0 \quad (4-15)$$

Substituting eq. (3-115) for P_e^* and eq. (3-11) for P_{ec}^* gives:

$$\frac{m^{*4}}{P_{q2}^{*2}} = -\frac{a}{3} (t_{out} - t_{in,o}) \frac{P_{e,o}^*}{P_{ec,o}^*} \frac{1 - \frac{P_{ec,o}^*}{P_{q2}^*} m^{*3}}{P_{q1,o}^* + P_{e,o}^* (1 + a(t_{out} - t_{out,o}))} \quad (4-16)$$

which may without great error be simplified to:

$$\left[\frac{m^{*2}}{P_{q2}^*} \right]^2 = -\frac{a}{3} (t_{out,o} - t_{in,o}) \frac{P_{e,o}^*}{P_{ec,o}^*} \frac{1}{1 + P_{e,o}^*} \quad (4-17)$$

For the numerical example of figs. 4-1 and 4-2 we have the following values of constants:

$$a = -0.005^{\circ}\text{C}^{-1}$$

$$t_{\text{out}} = 90^{\circ}\text{C}$$

$$t_{\text{in}} = 58.93^{\circ}\text{C}$$

$$P_{e,o}^* = 0.5$$

$$P_{ec,o}^* = 0.005$$

Inserting these values in eq. (4-17) gives at $P_{q2}^* = 1$:

$$m^* = 1.36$$

in accordance with the fig. 4-2.

From the approximate formula eq. (4-17) the following two general rules may be formulated:

- For a given system operating at different loads, to maximize the net electric output P_{e2}^* , the DH mass flowrate m^* should vary proportionally to the square root of the heat load.
- When the dimensions of the DH system are changed, so that the flow resistance of the system varies, then the DH flowrate m^* should be changed in proportion to the inverse of the fourth root of the pumping power $P_{ec,o}^*$ at reference flowrate.

According to Munser [1-1], the Russian Sokolov suggested that the relative value κ of the DH flowrate be varied with the relative value β of the heat load according to the empirical relation:

$$\kappa = \beta^m \quad (4-18)$$

where the exponent m is a constant to be determined. It is seen that eq. (4-17) derived above corresponds to $m = 0.5$. This result is essentially consistent with an equation for optimal flow variation developed in [1-37] for a DH system supplied with heat from a back-pressure cycle with regenerative feedwater heating. However, in [1-1] diagrams are given for optimal m as the result of various economic optimization calculations, where m takes all values between 0 and 1.

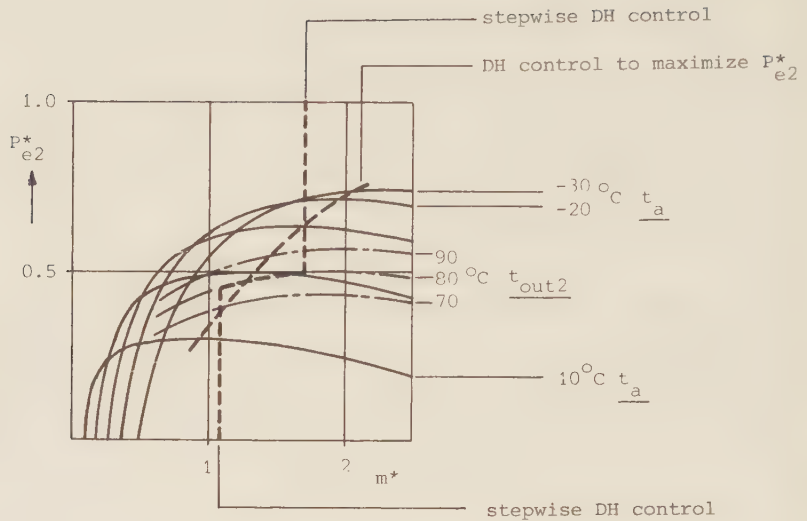


Fig. 4-4 Stepwise DH control according to Torisson [1-23].
Constant mass flowrate at high and low heat loads; constant supply temperature at medium loads.

Fig. 4-4 shows a different type of DH network control strategy, suggested by Torisson [1-23]. Here, at medium heat loads, the supply temperature is kept constant, while at higher or lower loads the flowrate is kept constant instead. This discontinuous type of control is seen to coincide with our previously derived strategy of maximum net electric output at three heat loads. At all other loads, Torisson's strategy is slightly inferior with regard to maximum P_{e2}^* .

A typical DH practice in Sweden for network control is to have a constant supply temperature at lower heat loads (typically above 0°C of ambient temperature), while at higher loads both the supply temperature and the

flowrate are raised with increasing heat load. The type of strategy we have arrived at in fig. 4-2 is thus roughly in accordance with this practice at the higher heat loads, whereas there is a discrepancy at lower loads. This difference is a result of the model simplification to exclude hot consumption water production, normally present in practical systems. The more refined model, discussed in Chapter 6, will remove this discrepancy.

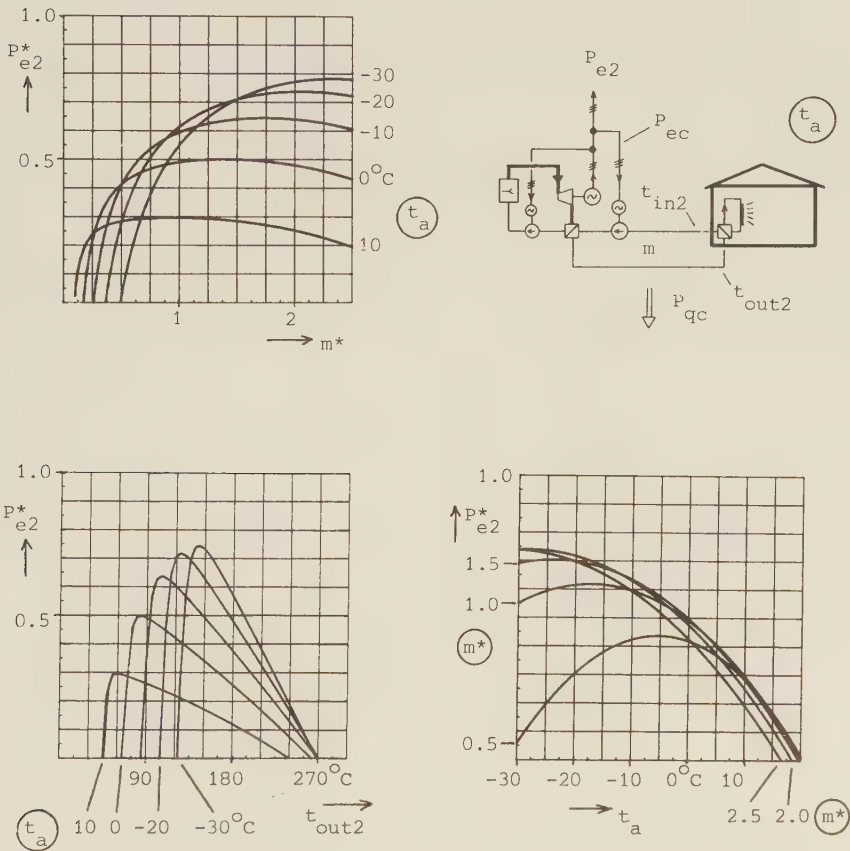
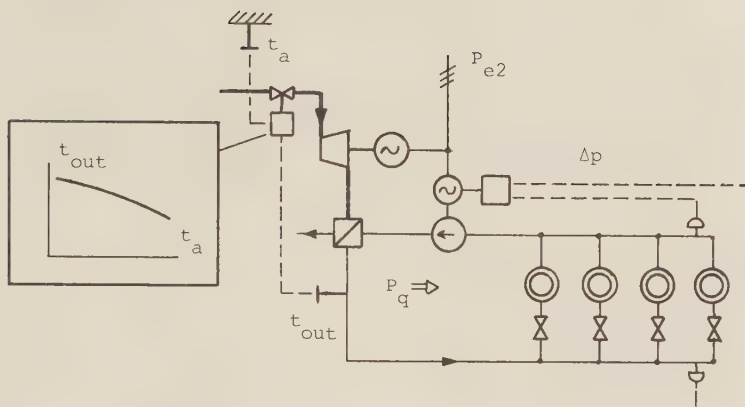


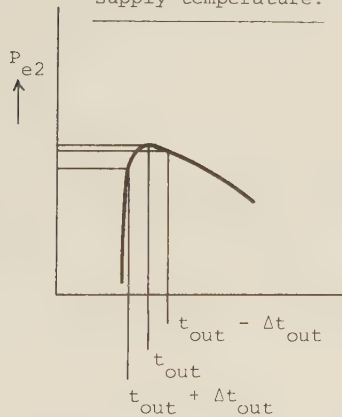
Fig. 4-5 Three ways of presenting results for net electric output for various ambient temperatures and various operational modes.

So far we have shown the load performance of our system with the mass flowrate m^* on the abscissa. However, in DH practice the control of the network is usually specified in terms of supply temperature instead of flowrate. In fig. 4-5 three types of presentation of load performance with P_{e2}^* on the ordinata have been put together. It can be seen that choosing t_{out2} for the abscissa instead of m^* results in much less symmetrical curves. At high temperatures the curves approach straight lines, because of the linear influence of condensing temperature t_c (not varying very differently from t_{out2}) on P_e^* , the electric output from the turbine. In the third diagram of fig. 4-5 the envelope connects points of maximum P_{e2}^* at each heat load. In this diagram the previously discussed global maximum for P_{e2}^* (when both heat load and mass flowrate are varied) is seen quite distinctly.

In fig. 4-6 the type of representation with the supply temperature on the abscissa has been used to discuss consequences of faults in a typical feedback control system in a DH network. This representation is appropriate here, since the turbine governing valve is controlled, not on the DH mass flowrate, but on the supply temperature. For simplicity differences between supply temperatures t_{out2} at the consumers and t_{out1} , measured at the production plant, have been neglected in the figure. The DH supply temperature turbine valve control is combined with a speed control of the circulation pump, holding the differential pressure between supply and return lines at the end point of the system. Thus at each load condition the DH mass flowrate automatically takes the value corresponding to the specified supply temperature and differential pressure. While the differential pressure is normally kept constant, the supply temperature can be varied in accordance with the ambient temperature, which, as the figure shows, is measured and transmitted to the control box of the valve. In the side-branches to the individual consumers, primary side control valves are inserted in series with heat exchangers. Their settings may be adjusted automatically according to each consumer's individual heat demand. Due to variations in demands from various consumers the total heat demand may deviate from the stipulated value, calculated from the ambient temperature. Thus, at a given ambient temperature there may be variations in the total mass flowrate to satisfy the condition of the fixed differential pressure at the end of the system. The reader may be referred here to the discussion in section 2.4 on effects of divided and distributed heat load.



Fault in setting of
supply temperature:



Fault in heat load:

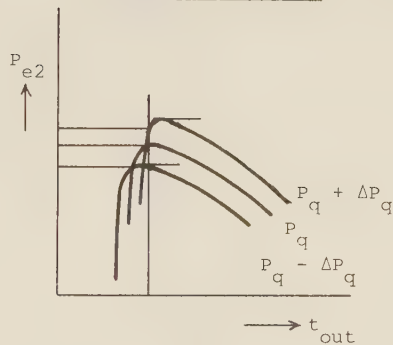


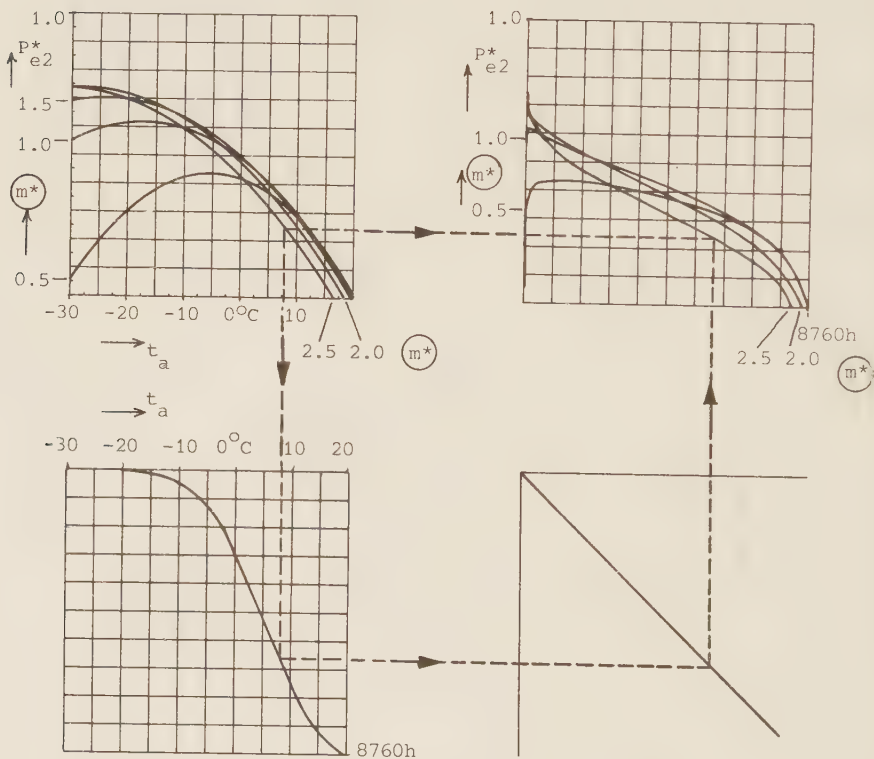
Fig. 4-6

Consequences for net electric output P_{e2} of two types of fault in DH control.

Fig. 4-7 Construction of duration curves for net electric output.

net electric output vs.
ambient temperature

duration curves for
net electric output



duration curve for
ambient temperature

In the left section of fig. 4-6 consequences are shown of two types of fault in control setting. The top diagram illustrates the case of a fault in the setting of t_{out} , e.g. resulting from a variation in the measurement of t_{out} . When such a variation causes the operation condition of the system to shift to the right of the point maximizing P_{e2}^* , then the loss of net electric output is only small. But, due to the asymmetry of the curve, an equally large shift to the left in the diagram has a significant adverse effect on P_{e2}^* . Thus in practice it will be wise to set the control on a supply temperature to the right of the point maximizing P_{e2}^* , i.e. with a marginal of safety against serious increases in pumping power. The diagram at the bottom of fig. 4-6 illustrates the case of variation in the heat load of the system at a given supply temperature. The ambient temperature, determining the supply temperature in the control system of the figure, is no more than an approximate measure for the actual aggregate heat demand. The demand is influenced by a number of additional factors, such as the wind speed. Again, control faults giving too low a value of the supply temperature may have a serious effect upon the net electric output, so that this type of fault also speaks in favour of a higher setting of the supply temperature than the one which in theory maximizes P_{e2}^* .

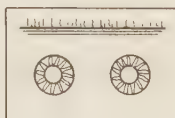
In addition to the net electric output at each heat load of a DH system with a back-pressure plant, the total output of electricity over a year will be of interest. This quantity can be calculated if the load duration curve is known. Fig. 4-7 shows how the $P_{e2}^* - t_a$ diagram from fig. 4-5 may be transformed into a duration curve diagram for net electric output from a load duration curve. The net output of electric energy may be calculated by integrating the resulting duration curves for P_{e2}^* . Instead of the integrated electric output one may calculate the integrated economic value of the output over a year. In cases where the value of each kW is not the same at all times of the year, as is frequently the case, such an integration may be a more relevant economic measure.

4.2 Technological Constraints of District Heating System

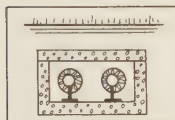
In the preceding section, various operational modes of a DH system were examined, allowing the water flowrate to vary freely. However, in practical systems a number of technological boundaries will restrict the range of operation. Thus, pressure drops are limited, both by maximum allowable stresses in the walls of medium-carrying pipes, and by the condition that at no point in the system should water pressure go below the boiling pressure. The maximum medium temperature will be limited by allowable axial expansion of pipes, or for other reasons.

When analysing such technological boundaries it is essential to discriminate between different types of pipeline design. For instance, when we discuss below how limitations on water pressure affect the DH operation range, we shall first examine the case of steel as a material for the medium-carrying pipe, and then the case of plastic pipes. In the first case allowable wall stresses are determined by the ductile limit of the material, in the second by creep rupture.

Commonly encountered pipeline designs are either of the concrete conduit type, or re-insulated steel pipes, see fig. 4-8. Pipes in concrete conduit



Separately insulated pipes with separate outer plastic shells.



Separately insulated pipes in a common concrete duct.

Fig. 4-8

Two types of DH pipeline design referred to in the text of sect. 4.2.

Further types of design are shown in fig. 3-4.

are allowed to expand at thermal loading, and deformations are taken up in compensators, placed at intervals along the pipes. The insulation material, usually being mineral wool, is capable of withstanding temperatures widely in excess of those in question for the water. A pre-insulated steel pipe may be designed, either to expand freely, with compensators, or it is restrained in its thermal expansion by a fixed connection between the steel pipe, a rigid insulation material, a plastic shield pipe, and the surrounding soil. Here, the insulation material may be exposed to creep loading, and in the boundary zone between the outer pipeshell and the soil the coefficient of friction is influenced by a number of factors, including the variable moisture content of the soil. Thus, with such a design the limit medium temperature is less easily determined.

The various types of pipeline design have reached different degrees of technological maturity. The concrete conduit type, having been in use for several decades, may be designated as a safe, but also rather expensive design. Pre-insulated pipes, due to their greater simplicity, are cheaper and have in the last decade reached a widespread application, above all in Scandinavia [4-1], while DH practice in the FRG has been more conservative on this point [4-2]. Although many innovations, claiming to improve operation reliability of various types of pre-insulated pipe designs have been introduced, and systematic statistics on field experience with the different designs have been reported [4-1], general consensus on the reliability question does not exist. Pipeline designs with plastic medium-carrying pipes instead of steel have been investigated by research institutes, [4-3], [4-4], and have recently been introduced on the market, but for those designs systematic and prolonged field tests are even more lacking.

In addition to internal loading of pipelines from water pressure, temperature, etc, various types of external loads may be present. Examples are soil pressures from variations in ground water table, and corrosion because of water transported to the outer surface of medium-carrying pipes. As Brachetti has pointed out [1-19], a consequence of simplified pipeline designs, such as pre-insulated pipes, is that each element of the design must carry out more types of functions than in the case of the traditional concrete-duct design. For instance, medium-carrying pipes may be exposed to, not only forces from the flowing medium, but also mechanical loads from the soil. The greater the degree of such integration of functions, the greater is the

risk that internal and external loads may interact in various types of damage mechanisms. External pipe corrosion is commonly involved in the damage picture of a DH pipeline failure. Although in the case of a pre-insulated pipe without compensators effective drainage of the soil will be an important measure in preventing such corrosion, a reduced medium temperature may also contribute to reducing the risk of corrosion.

Reports from practical operation with DH networks seem to indicate that pipeline damage frequency tends to increase at variations in operation conditions. Thus, a sound strategy for the operation of the network may be to let the supply temperature follow variations in heat load only to a limited degree, neglecting fast fluctuations in load, when setting the supply temperature.

Complicated damage mechanisms, like those just mentioned, call for a modest attitude when attempting to set limits for DH system operation on purely deterministic grounds. Nevertheless, as we shall see, part of the question may be treated successfully with rather simple theoretical tools.

First we consider the combined effects of pipe strength limit and of the risk of DH system water boiling. For simplicity we shall suppose that the mechanical loading of the pipes is always greatest (in terms of actual stress vs. rupture stress) in the supply pipe. This is in accordance with construction practices, where equal pipe wall thickness are applied in supply and return lines. The normally lower operation pressure in return pipes might suggest that they be designed for a smaller wall thickness. Such a practice would, however, hardly be acceptable, since one can never guarantee that full supply line pressures may not in some situations be transferred to the return line, e.g. through consumers' connections.

Fig. 4-9 shows how a diagram for permissible operation range of the mass flow m^* at different ambient temperatures may be constructed. The top left diagram gives the variation in supply and return line pressures at the circulation pump. The pressure in the supply line is taken to be constant at load variations, controlled by the pressure holding system of the network. The pressure in the return line then goes down with increasing m^* , the differential pressure varying approximately with the square of the mass flow, as discussed in section 2.4. A constant maximum supply pressure means that

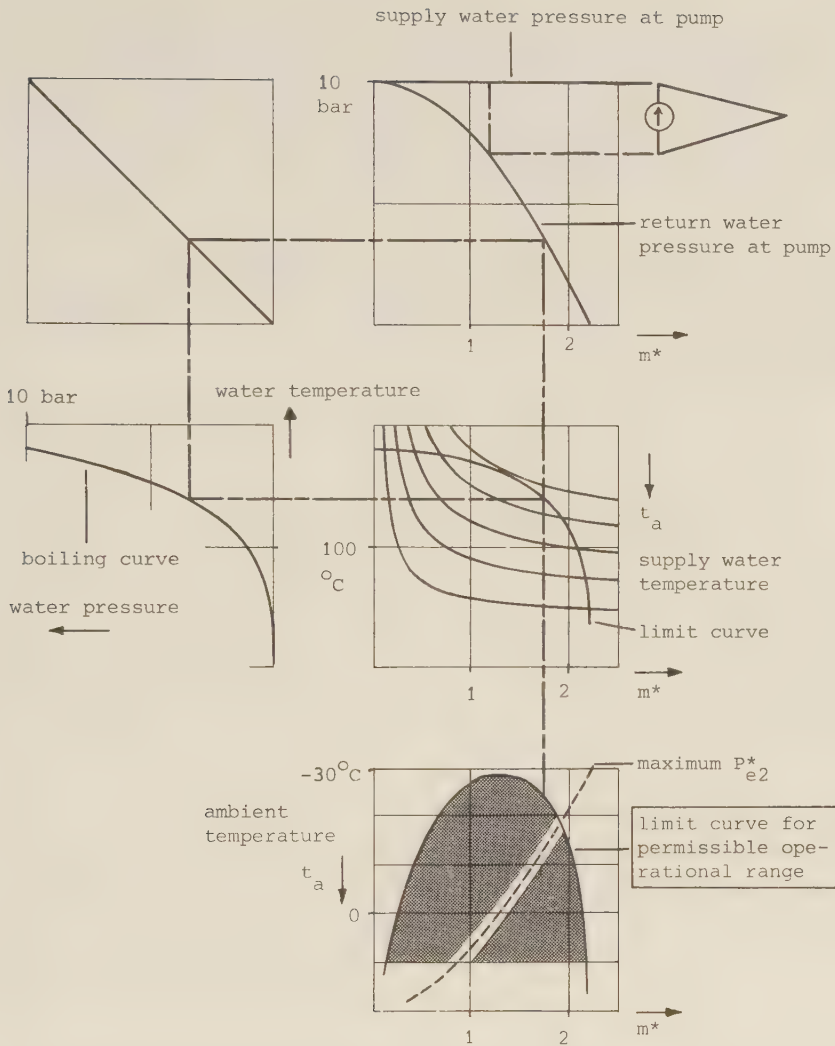


Fig. 4-9

Construction of permissible operational range for DH water mass flowrate to avoid boiling in return line at pump in case of water overflow. Pressure in supply line determined by ductile material strength limit.

the strength margin for the supply pipe is constant, provided the strength is determined by ductile material flow, as can be assumed for normal steel pipe material. If instead the maximum supply pressure was allowed to vary with the mass flow (e.g. if the mean pressure between supply and return is fixed), then the resulting operation range for m^* would be smaller.

At the top middle of fig. 4-9 is shown the variation of the supply temperature (taken at the circulation pump) with mass flow m^* and ambient temperature t_a , and at the bottom the boiling curve for water. The boiling criterion applied here is this:

- No boiling should occur to overflow water from the supply to the return line at the circulation pump location, even if the overflow water is not cooled at all when experiencing the full pressure drop from supply to return.

This criterion is conservative since water passing from the supply to the return is usually cooled, either when passing a heat exchanger, or when mixing with water in the return line. Still, cases may occur when this is not so. For avoiding cavitation in valves, an even stricter criterion than the above may be necessary.

Combination of the top left pressure diagram and the boiling curve gives the limit curve in the temperature vs. mass flow diagram. The straight line at 45° inclination assists in the graphical construction. The limit curve connects points where boiling according to the criterion above sets in. For a given ambient temperature the limit values for m^* are found at intersection points with the limit curve. From this the right-hand diagram of m^* vs. t_a has been constructed.

It may be noted that for each t_a there is both a maximum and a minimum m^* . When m^* is increased beyond its maximum, it is true that the supply temperature goes down, which in itself is favourable in relation to the boiling criterion, but at the same time the pressure in the return line (at the pump) drops dramatically, so that boiling nevertheless occurs. Similarly, diminishing m^* below its minimum value gives an increase in supply temperature which is so large that it is not compensated for by the relatively high return line pressure.

In the resulting diagram for m^* is also shown the previously developed curve for m^* corresponding to maximum net electric output from a back-pressure plant. It can be seen that at -18°C , in the present numerical example, here this curve intersects with the upper branch of the limit curve for m^* . Thus, if the ambient temperature goes below this value, a departure from the desired values for m^* must be accepted. Due to the moderate variation in P_{e2}^* with m^* around maximum, the loss in P_{e2}^* resulting from this departure may be small. In any case the best choice will be to follow the upper branch of the limit curve at high loads. To diminish the mass flow m^* with increasing (large) heat loads may be considered contrary to DH practice. One could therefore suspect that a number of DH systems in practice have built-in reserves of heat transmission capacity which the operators are not fully aware of.

The boiling criterion based on water overflow from supply to return at the circulation pump can be applied to the case of a branched DH system with many consumers. Here the greatest risk exists for the consumer, who is located where the pressure in the return line is minimum, i.e. usually at short distance from the circulation pump. With networks designed for hilly areas, the lowest return line pressure may be found at a consumer at high altitude. A different situation exists when the pipes from the circulation pump are transmission lines with no connection between the supply and return. In that case the boiling criterion must be modified to refer to the actual point of maximum boiling risk, presumably at the nearest consumer, as seen from the circulation pump. If instead the one circulation pump shown in fig. 4-9 is divided into two or more pumps, at different locations in the system, more points of the DH system must be considered as important when assessing the boiling risk.

The pressure distribution shown in fig. 4-9 assumes that no variations in geodetic level are present. In case of such variations, points where there is the greatest risk of boiling may be found away from the circulation pump.

In fig. 4-9 the maximum pressure in the supply line is determined by ductile material flow. Fig. 4-10 shows an analogous construction of permissible range for m^* , when the supply pipe is exposed to material creep instead, such as would be the case with a pipe wall material of plastic. It is still a debatable matter how to predict with reasonable confidence the

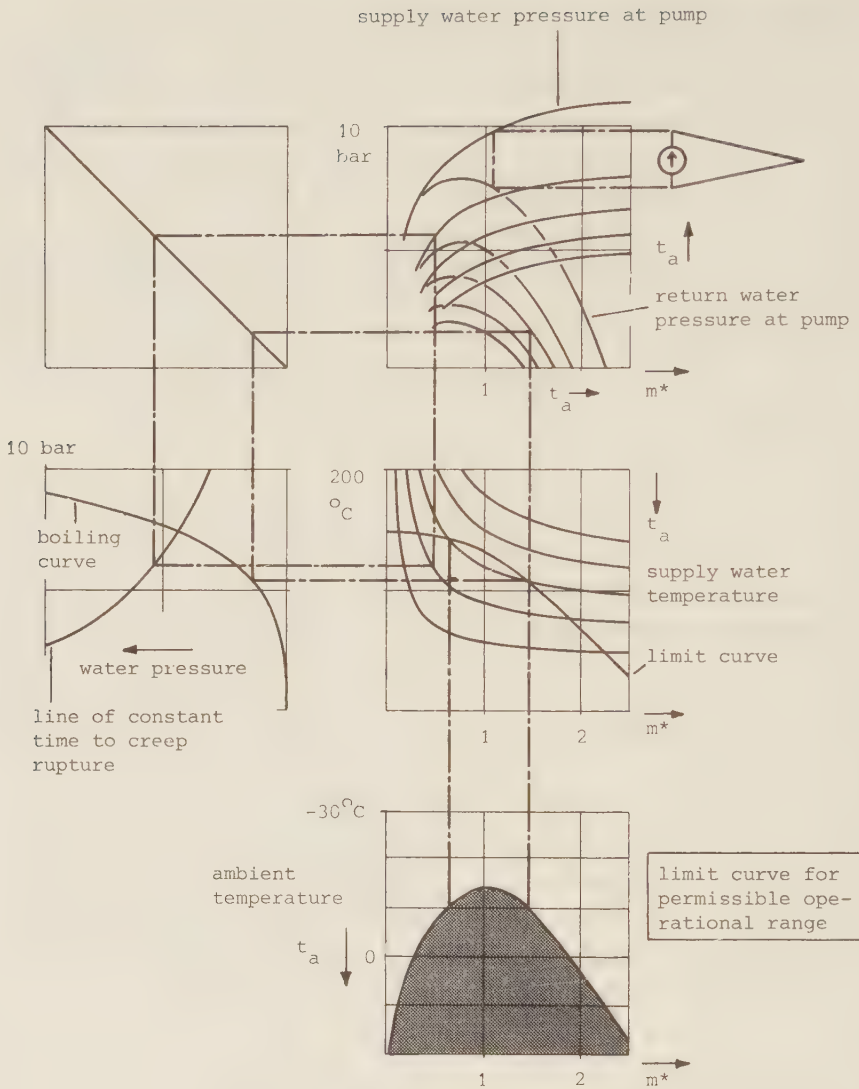


Fig. 4-10

Construction of permissible operational range for DH water mass flowrate to avoid boiling in return line at pump in case of water overflow.

Pressure in supply line determined by creep rupture strength limit.

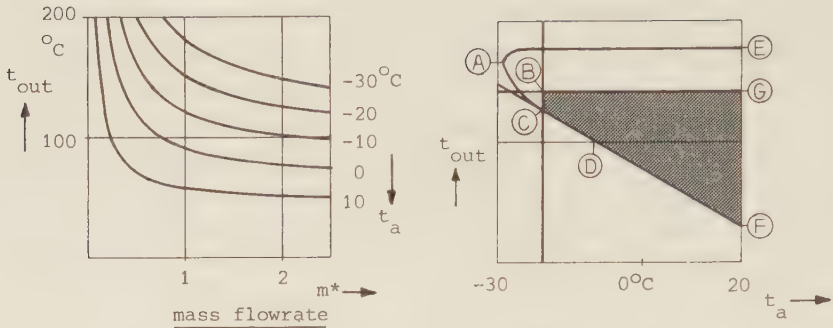
technical lifetime with field operation of such pipes. One possibility is to prescribe that for each value of water temperature in the supply line, the pressure may be selected corresponding to a given expected lifetime from creep rupture curves, selecting the same lifetime for all values of the temperature. A curve connecting such points has been indicated in the diagram of fig. 4-10 also giving the boiling curve for water. The resulting variation of supply and return pipe pressures are no longer two curves only, as in fig. 4-1, but a family of curves. The range of m^* in the diagram to the right in fig. 4-10 is similar in form of that of fig. 4-9, though in the numerical example here the absolute minimum for t_a has shifted to a higher value.

In fig. 4-11 are shown the effects on the permissible DH operation range of various types of technological boundaries, in addition to those of fig. 4-9. Two alternative types of graphical representation are shown. At the bottom the ordinata is the DH mass flowrate as before, and at the top it is the DH supply temperature, more commonly specified in DH practices. The temperature-mass flow diagram in the upper left-hand corner can be utilized to transform one of the two operation boundary diagrams into the other.

The line B-C in the diagrams is not a technological boundary deriving from the DH system proper, but to the internal heating system of the building(s). This minimum ambient temperature may be determined according to various criteria. As an ultimate demand, the radiator water should not be allowed to boil, i.e. for atmospheric pressure of the water, the temperature should not exceed 100°C . However, hygienic demands (burning of dust particles) and safety considerations (risk of burning accidents) may impose a lower setting of the maximum radiator water temperature.

It can be seen from fig. 4-11 that sometimes one type of technological limit may make another limit ineffective. For instance, in the example shown the supply temperature limit is set so high that the lower part of the parabola-like curve in the bottom diagram is not part of the boundary curve for the operation range. Also, had the limit temperature given by the vertical B - C been set some degrees higher, no part of the parabola-like curve would have been included. In that case it may be said that the DH system and the internal radiator heating system are not too well matched. When for a DH system some kind of changed operation, e.g. lower supply temperature, is

DH supply temperature:



- A - E: Minimum m^* to avoid boiling in return pipe at overflow
- C - D: Maximum m^* to avoid boiling in return pipe at overflow
- B - G: Minimum m^* to avoid exceeding maximum t_{out}
- D - F: Maximum m^* to avoid erosion, noise, etc.
- B - C: Limit given by maximum heating capacity of building radiator system

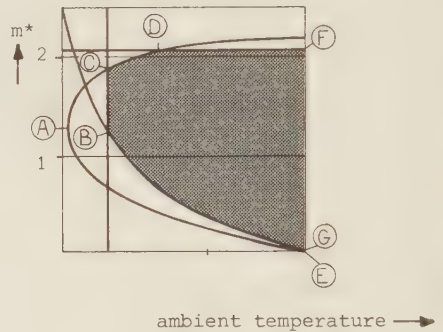


Fig. 4-11

Permissible operational range for DH supply temperature t_{out} and for DH mass flowrate m^* . Effects of various types of technological restraints indicated.

considered, it would be appropriate to examine matchings of different boundaries, to see where possible modifications in system design are most rationally made.

When designing an internal heating system for a building the engineer must select sizes of radiator surfaces in accordance with the climatic conditions of the particular geographical region, and with the insulation standard of the building. It is standard procedure to operate with the concept of ambient design temperature, t_{aD} , which is the lowest ambient temperature at which the heating system is supposed to be capable of maintaining the prescribed room temperature. At such heat load conditions, the radiator surface temperature reaches its maximum permissible value. t_{aD} is stipulated in national design codes, which in turn are based upon statistical climatic data.

If, for a given building, with a given insulation standard, the design ambient temperature is altered, i.e. if the sizes of radiator surfaces are changed, then the functional relationship between ambient temperature and radiator temperature will change accordingly. The top left-hand diagram of fig. 4-12 shows how the net electricity production, as a function of the ambient temperature, will shift, when such a change is made for the radiator system. The curves may also be taken as valid for the case of an unchanged radiator system, but variable insulation standard. If, for instance, the insulation thickness for a given building is increased, then the unchanged radiator heating system becomes larger in relation to the smaller heat load. Thus, the ambient design temperature is being artificially lowered, and the system performance is shifted to a curve more to the left of the diagram. Although this curve is below the original one, the electric output of the back-pressure plant does of course not deteriorate at a given ambient temperature, since the ordinata in the diagram is the dimensionless net electric output, made dimensionless by different reference heat loads in the two cases compared.

The family of curves in the top left-hand diagram unify to a single curve when, as shown in the diagram to the right, the heat load P_{q2}^* is substituted for t_a on the abscissa. Below that diagram is shown an additional scale, for which the heat load is divided by, not the reference heat load $P_{q2,0}$, as with P_{q2}^* , but the heat load of P_{q2D} at design ambient temperature. In the numerical example of the figure P_{q2D} is seen to be twice $P_{q2,0}$. They

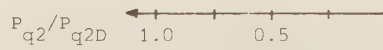
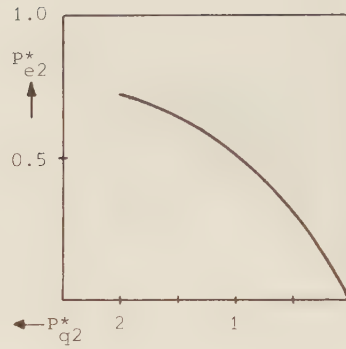
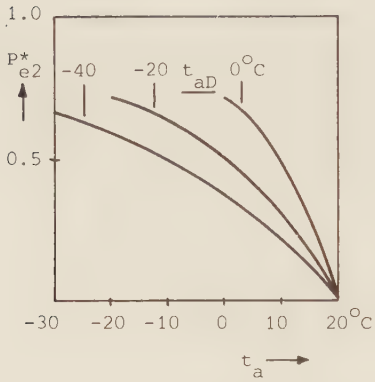
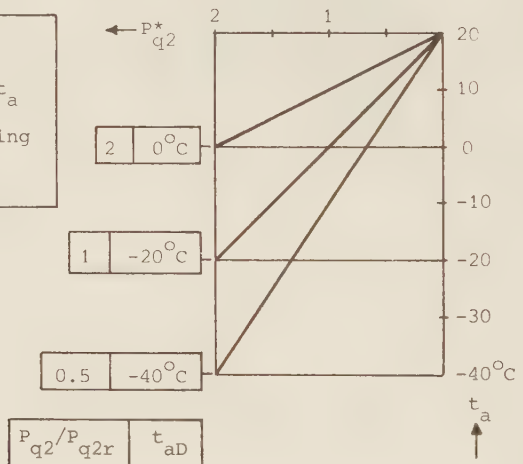


Fig. 4-12

Substituting P_{q2}^* for t_a as abscissa to unify curves for P_{e2}^* referring to different building insulation standards.



P_{q2}^* = dimensionless heat load as used elsewhere in the text

P_{q2D} = heat load at ambient design temperature

P_{q2r} = size of heat load when standard insulation applied, heat load varying with ambient temperature t_a

t_{aD} = ambient design temperature

would be identical, had t_{aD} been selected as the reference temperature, $t_{a,O}$. Below is shown a diagram, which displays the relationship between dimensionless heat loads of the diagram above, and ambient temperature. The straight lines here refer to different ambient design temperatures and varying building insulation standards.

The top diagram may be used, not only to compare cases of different radiator surface sizes, or different insulation standards for a given DH system with given buildings, but also for comparison of DH systems in different climatic regions. Naturally, such systems must then have equal constants in the systems of equations constituting the thermodynamic system. For instance, the heat exchangers must be identical. A slight theoretical difficulty here is the fact that the complete equivalence of such systems being compared, requires that the soil temperatures (influencing DH heat losses) must be equal, an assumption which is not in accordance with reality, where the soil temperature varies with the climatic region. However, if the insulation thicknesses of the DH pipelines are changed in relation to the soil temperature, the systems will be virtually equivalent.

4.3 Variation of System Dimensions

In section 4.1 various operational modes for our simple system model were analysed. The results obtained there will be of interest to the control room engineer, who is capable of influencing the DH supply water temperature. Normally he will not have any possibility of changing system dimensions. Such variation of parameters will instead be of relevance to design engineers and decision makers responsible for design and modification of DH system.

In accordance with the intentions expressed already in section 1.1 (Focus of Interest), the investigation below will concentrate on showing consequences of varying parameters of the DH system proper, while for instance changes in radiator surface area are not considered. We shall examine changes in pipe insulation, in pipe length and diameter, and in dimensions of heat exchangers, both at the generating plant and at the consumer's connection.

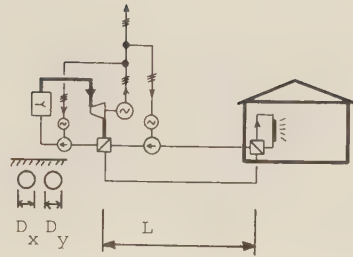
From the way we shall define those parameters in the following it will be noted that our investigations will specialize to some few of the many DH system design parameter variations which are possible. Part of the specialization is a consequence of the fact that our model has only a single heat consumer, instead of many consumers served by a branched network, as was discussed in section 2.4. In general, the results derived in this chapter will need a more elaborate interpretation than was the case for results from pure load performance calculations in section 4.1.

Derivation of equations

Table 4-4 summarizes the system of equations utilized below when calculating consequences of changes in system dimensions. The equations are here grouped into three blocks, which at computer calculations are run through in consecutive order. The last of the blocks, giving values of system parameters at changed operational conditions, has been taken over unchanged from table 4-1. The block of equations in the middle yields parameter values at changed system dimensions and reference operational mode ('starting values' for the calculations in the last block). Finally, the equations in the first block are necessary for the calculation of the values of a few parameters at both reference system dimensions and reference operational mode.

Table 4-4

System of equations for calculating consequences of variations in system dimensions.



Dimensionless variables

Pipe diameter:	$D^* = D_x/D_{x,r} = D_y/D_{y,r}$
Pipe length:	$L^* = L/L_r$
Heat exchanger length:	$L_1^* = L_1/L_{1,r}, L_2^* = L_2/L_{2,r}$
Heat transfer surfaces:	$S_1^* = S_1/S_{1,r}, S_2^* = S_2/S_{2,r}$
Pipe insulation numbers:	$I_{IO}^*, I_{II}^*, I_{III}^*$

Calculation of variable values at reference system dimensions (index r) and at reference operational mode (index o):

$$w_r = 1 - x_r - y_r - z_r$$

$$t_{in2,or} = t_{out2,or} - (t_{out2,or} - t_{1,o}) (1 - \exp(-U_{2,or}^*))$$

$$P_{qc,or}^* = P_{qIO,or}^* + P_{qII,or}^*$$

$$P_{q1,or}^* = 1 + P_{qc,or}^* - P_{ec,or}^*$$

$$t_{in1,or} = t_{in2,or} + (t_{out2,or} - t_{in2,or}) \frac{y_r P_{ec,or}^* - P_{qII,or}^* + P_{qIII,or}^*}{1 - z_r P_{ec,or}^*}$$

$$t_{out1,or} = t_{in1,or} + (t_{out2,or} - t_{in2,or}) \frac{P_{q1,or}^* + w_r P_{ec,or}^*}{1 - z_r P_{ec,or}^*}$$

$$t_{c,or} = t_{in1,or} + (t_{out2,or} - t_{in2,or}) \frac{(P_{q1,or}^* + w_r P_{ec,or}^*)}{1 - \exp(-U_{1,or}^*)} \frac{1/(1 - z_r P_{ec,or}^*)}{1 - \exp(-U_{1,or}^*)}$$

(to be continued)

$$P_{ec,o}^* = P_{ec,or}^* (w_r L_1^3 S_1^{*-2} + z_r L_2^3 S_2^{*-2} + (x_r + y_r) L^* D^{*-5}) \quad (4-32)$$

$$x = x_r L^* D^{*-5} P_{ec,or}^* / P_{ec,o}^* \quad (4-25) \quad y = y_r L^* D^{*-5} P_{ec,or}^* / P_{ec,o}^* \quad (4-26)$$

$$z = z_r L_2^3 S_2^{*-2} P_{ec,or}^* / P_{ec,o}^* \quad (4-30) \quad w = w_r L_1^3 S_1^{*-2} P_{ec,or}^* / P_{ec,o}^* \quad (4-31)$$

$$U_{2,o}^* = U_{2,or}^* L_2^* S_2^{*1-p_2} \quad (4-36)$$

$$t_{out2,o} = t_{1,o} + (t_{out2,or} - t_{1,o}) \frac{1 - \exp(-U_{2,or}^*)}{1 - \exp(-U_{2,o}^*)}$$

$$t_{in2,o} = t_{in2,or} + t_{out2,o} - t_{out2,or}$$

$$j = I_{IO}^* L^* D^* P_{qIO,or}^* / (t_{out2,or} - t_s)$$

$$P_{qIO,o}^* = I_{IO}^* L^* D^* P_{qIO,or}^* + j(t_{out2,o} - t_{out2,or})$$

$$k = I_{IIO}^* L^* D^* P_{qIIO,or}^* / (t_{in2,or} - t_s)$$

$$P_{qIIO,o}^* = I_{IIO}^* L^* D^* P_{qIIO,or}^* + k(t_{in2,o} - t_{in2,or})$$

$$l_{10} = I_{III}^* L^* D^* P_{qIII,or}^* / (t_{out2,or} - t_{in2,or})$$

$$l_{01} = -l_{10}$$

$$P_{qIII,o}^* = I_{III}^* L^* D^* P_{qIII,or}^* + l_{10}(t_{out2,o} - t_{out2,or}) + l_{01}(t_{in2,o} - t_{in2,or})$$

$$P_{qc,o}^* = P_{qIO,o}^* + P_{qIIO,o}^*$$

$$P_{q1,o}^* = 1 + P_{qc,o}^* - P_{ec,o}^*$$

$$t_{in1,o} = t_{in2,o} + (t_{out2,o} - t_{in2,o}) \frac{y P_{ec,o}^* - P_{qIIO,o}^* + P_{qIII,o}^*}{1 - z P_{ec,o}^*}$$

$$t_{out1,o} = t_{in1,o} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1,o}^* + w P_{ec,o}^*}{1 - z P_{ec,o}^*}$$

$$U_{1,o}^* = U_{1,or}^* L_1^* S_1^{*1-p_1}$$

$$t_{c,o} = t_{in1,o} + (t_{out2,o} - t_{in2,o}) (P_{q1}^* + w P_{ec,o}^*) \frac{1/(1 - z P_{ec,o}^*)}{1 - \exp(-U_{1,o}^*)}$$

$$m_{s,o}^* = P_{q1}^* / (P_{q1,or}^* (1 + b(t_{c,o} - t_{c,or})))$$

$$P_{e,o}^* = P_{e,or}^* m_{s,o}^* (1 + a(t_{c,o} - t_{c,or}))$$

$$P_{eb,o}^* = P_{eb,or}^* m_{s,o}^* (1 + e(t_{c,o} - t_{c,or}))$$

Table 4-4, continued

Calculation of variable values at changed system dimensions and at reference operational mode (index o).

(to be continued)

$$P_{q2}^* = 1 - i(t_a - t_{a,o})$$

$$t_1 = t_h + (t_{1,o} - t_h) P_{q2}^{*1/n}$$

$$P_{ec}^* = P_{ec,o}^* m^{*3}$$

$$U_2^* = \frac{U_{2,o}^*}{m^*} \frac{1 + x_2}{m^{*-P_2} + x_2}$$

Table 4-4, continued

Calculation of variable values
at changed system dimensions
and at changed operational mode.

$$t_{out2} = t_1 + (t_{out2,o} - t_{1,o}) \frac{P_{q2}^* - zP_{ec}^*}{1 - zP_{ec,o}^*} \frac{1}{m^*} \frac{1 - \exp(-U_{2,o}^*)}{1 - \exp(-U_2^*)}$$

$$t_{in2} = t_{out2} - (t_{out2} - t_1)(1 - \exp(-U_2^*))$$

$$P_{qIO}^* = I_{IO}^* L^* D^* P_{qIO,or}^* + j(t_{out2} - t_{out2,or})$$

$$P_{qII0}^* = I_{II0}^* L^* D^* P_{qII0,or}^* + k(t_{in2} - t_{in2,or})$$

$$P_{qIII}^* = I_{III}^* L^* D^* P_{qIII,or}^* + l_{10}(t_{out2} - t_{out2,or}) + l_{01}(t_{in2} - t_{in2,or})$$

$$P_{qc}^* = P_{qIO}^* + P_{qII0}^*$$

$$P_{q1}^* = P_{q2}^* + P_{qc}^* - P_{ec}^*$$

$$t_{in1} = t_{in2} + (t_{out2} - t_{in2}) \frac{yP_{ec}^* - P_{qII0}^* + P_{qIII}^*}{P_{q2}^* - zP_{ec}^*}$$

$$t_{out1} = t_{in1} + (t_{out2} - t_{in2}) \frac{P_{q1}^* + wP_{ec}^*}{P_{q2}^* - zP_{ec}^*}$$

$$U_1^* = \frac{U_{1,o}^*}{m^*} \frac{1 + x_1}{m^{*-P_1} + x_1}$$

$$t_c = t_{in1} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1}^* + wP_{ec}^*}{m^*} \frac{1/(1 - zP_{ec,o}^*)}{1 - \exp(-U_1^*)}$$

$$m_s^* = m_{s,o}^* P_{q1}^* / (P_{q1,o}^* (1 + b(t_c - t_{c,o})))$$

$$P_e^* = P_{e,o}^* m_s^* m_{s,o}^{*-1} (1 + a(t_c - t_{c,o}))$$

$$P_{eb}^* = P_{eb,o}^* m_s^* m_{s,o}^{*-1} (1 + e(t_c + t_{c,o}))$$

$$P_{e1}^* = P_e^* - P_{eb}^*$$

$$P_{e2}^* = P_{e1}^* - P_{ec}^*$$

Table 4-5

Input values for parameters in table 4-3, valid for all results shown in diagrams in section 4.3.

Constants:

t_s	=	0°C
t_h	=	20°C
n	=	1.3
x_1	=	0.1
x_2	=	1.0
p_1	=	0.5
p_2	=	0.5
x_r	=	0.25
y_r	=	0.25
z_r	=	0.25
a	=	-0.005
b	=	0
e	=	-0.001
i	=	-0.05

Values defining reference conditions:

$t_{a,o}$	=	0°C
$t_{1,o}$	=	50°C
$t_{\text{out}2,\text{or}}$	=	90°C
$U_{1,\text{or}}^*$	=	1.5
$U_{2,\text{or}}^*$	=	1.5
$P_{q\text{IO},\text{or}}^*$	=	0.04
$P_{q\text{IIO},\text{or}}^*$	=	0.04
$P_{q\text{III},\text{or}}^*$	=	0.04
$P_{\text{ec},\text{or}}^*$	=	0.005
$P_{\text{eb},\text{or}}^*$	=	0.005
$P_{\text{e},\text{or}}^*$	=	0.5

As was stated in section 1.4 (Mathematical Formalism), index 'or' is being attached to a parameter to indicate that its value is taken at both reference system dimensions and at reference operational mode, while index 'o' (as before) designates reference operational conditions only.

The mathematical apparatus presented here will enable us to vary system dimensions and operational mode independently of each other. Thus, whenever some system dimension is being changed, we shall be free to choose the operational condition which may be appropriate. The importance of this feature of the equations will become clear when presenting numerical results later in the section.

Below the system schematic of table 4-4, those variables which define changed system dimensions, are summarized. As can be seen, the variables are given in dimensionless form, relating their absolute value to a reference value, indicated by index 'r'. Instead of 'r', index 'or' could have been used here, but since system dimensions do not change at variations in operational mode, the shorter index convention will be sufficient in this case.

The significance of the dimensionless pipe dimensions, D^* and L^* is readily understood, while the precise meaning of the other variables below the system figure will become clear from the explanations below. It may be noted that as a specialization we have chosen, by operating with a single dimensionless diameter D^* , to let supply and return pipes change in identical proportions.

When varying pipeline insulation thickness, it has been decided to choose as parameters, not insulation dimensions directly, but what is designated as dimensionless insulation numbers, I_{IO}^* , I_{IIO}^* , and I_{III}^* , whose meaning can be seen from the equations giving $j_{qIO,o}^{P*}$, $k_{qIIO,o}^{P*}$, l_{10} , and $P_{qIII,o}^{P*}$.

Heat loss components are seen to vary proportionally with changes in pipe surface area.

The first of the dimensionless insulation numbers is defined by the following equation:

$$I_{IO}^* = \frac{\frac{P^* q_{IO,o}}{L\pi D_x} t_{out2,or} - t_s}{\frac{P^* q_{IO,or}}{L\pi D_{x,r}} t_{out2,o} - t_s} \quad (4-19)$$

and analogously for the two other insulation numbers:

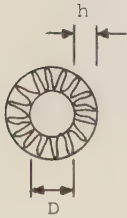
$$I_{IIO}^* = \frac{\frac{P^* q_{IIO,o}}{L\pi D_y} t_{in2,or} - t_s}{\frac{P^* q_{IIO,or}}{L\pi D_{y,r}} t_{in2,o} - t_s} \quad (4-20)$$

$$I_{III}^* = \frac{\frac{P^* q_{III,o}}{L\pi D_x} t_{out2,or} - t_{in2,or}}{\frac{P^* q_{III,or}}{L\pi D_{x,r}} t_{out2,o} - t_{in2,o}} \quad (4-21)$$

From these definitions it is seen that at large pipe diameters, dimensionless insulation numbers are inversely proportional to insulation thickness, provided the heat conductivity of the insulation material remains constant. But when the pipe diameters are not dominant in relation to insulation thickness, a distortion effect arises, cf. fig. 4-13. This can be illustrated by considering the case of changed diameters and lengths ($D^* \neq 1$ and $L^* \neq 1$), but unchanged insulation numbers ($I_{IO}^* = I_{IIO}^* = I_{III}^* = 1$) and heat conductivity. Here, insulation thicknesses must alter their values.

One advantage with the introduction of dimensionless insulation numbers is that by varying their values in calculations of system characteristics, effects of not only changed insulation thickness may be included, but also of changed heat conductivity of insulation material.

In order to arrive at an expression which relates DH pumping power $P_{ec,o}^*$ at changed system dimensions to the value $P_{ec,or}^*$ at reference system dimensions, we shall consider below first losses in pipelines, and then losses in heat exchangers.



Case of circular insulation:

Heat flow per square unit of inner pipe surface:

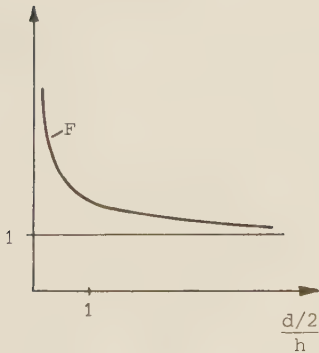
$$q''_c = \frac{\lambda \Delta t}{d/2} \frac{1}{\ln(1 + \frac{h}{d/2})}$$



Case of plane wall insulation:

Heat flow per square unit:

$$q''_p = \frac{\lambda \Delta t}{h}$$



Ratio between heat flows in the two cases shown above:

$$\frac{q''_c}{q''_p} = \frac{\frac{h}{d/2}}{\ln(1 + \frac{h}{d/2})} = F(\frac{h}{d/2})$$

Fig. 4-13 Comparison of heat losses through circular and plane wall insulations.

For the pressure drop over a pipeline we have the familiar expression:

$$\Delta p = \lambda \frac{L}{D} \frac{1}{2} \rho v^2 \quad (4-22)$$

By definition $m^* = 1$ when $P_{ec}^* = P_{ec,o}^* = P_{ec,or}^*$, so that we may write:

$$\frac{x P_{ec,o}^*}{x_r P_{ec,or}^*} = \frac{\Delta p_x}{\Delta p_{x,r}} = \frac{\lambda}{\lambda_r} L^* D^{*-1} \left(\frac{v}{v_r} \right)^2 \quad (4-23)$$

Assuming that conditions of fully turbulent flow prevail, and that at changes in pipe diameter the relative surface roughness remains constant, we have:

$$v/v_r = D^{*-2} \quad (4-24)$$

which inserted in eq. (4-23) gives:

$$x = x_r L^* D^{*-5} P_{ec,or}^* / P_{ec,o}^* \quad (4-25)$$

Analogously we may write for the return line:

$$y = y_r L^* D^{*-5} P_{ec,or}^* / P_{ec,o}^* \quad (4-26)$$

Both eqs. (4-25) and (4-26) are utilized in table 4-3.

We now turn to the heat exchangers, i.e. the turbine condenser and the heat exchanger at the consumer's connection. As the list of dimensionless system dimensions below the system figure of table 4-3 indicates, we shall allow both heat transfer surface areas to be varied, and what is being designated as 'heat exchanger lengths'.

The concept of dimensionless heat exchanger length will apply to various types of heat exchanger geometry. At the top of fig. 4-14 two common types of consumer connection heat exchangers are shown, the pipe heat exchanger and the spiral-tube exchanger with secondary water in cross-flow over

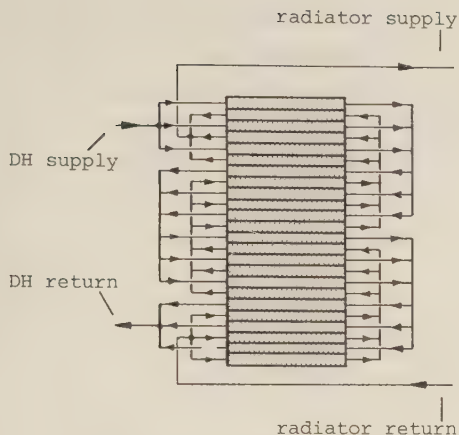


Fig. 4-14A

Schematic of plate heat exchanger with counterflow between DH water and radiator water of internal building heating system.

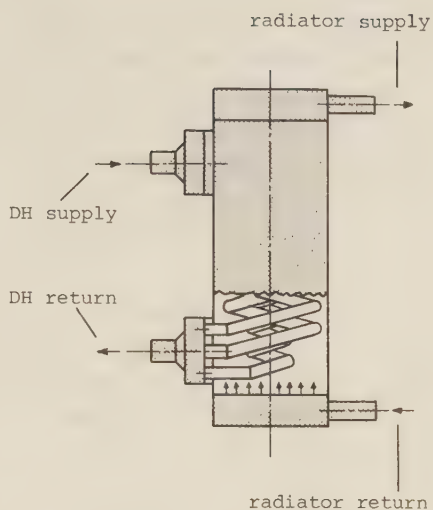


Fig. 4-14B

Spiral-tube heat exchanger with high-pressure DH water inside tubes. Radiator water in cross-flow over tube banks.

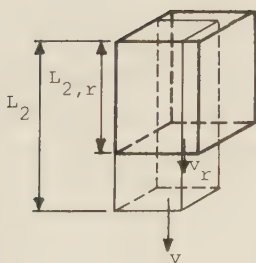


Fig. 4-14C

Theoretical schematic of change in heat exchanger length with constant heat transfer surface.

Dimensionless heat exchanger lengths:

$$L_1^* = L_1 / L_{1,r} \quad L_2^* = L_2 / L_{2,r}$$

Dimensionless heat transfer surfaces:

$$S_1^* = S_1 / S_{1,r} \quad S_2^* = S_2 / S_{2,r}$$

Indexes 1 and 2 refer to turbine condenser and to consumer's heat exchanger, respectively.

tube bundles. The heat exchanger length designates the length of the total flow path of primary water passing the heat exchanger surface. For both the heat exchanger types shown in the figure, this length will be greater than the external geometric length of the apparatus. The dimensionless heat exchanger length, then, is defined as the ratio between the variable absolute length L_2 and the corresponding reference length $L_{2,r}$ (in the case of the consumer's heat exchanger, indicated by the index 2).

Dimensionless heat exchanger lengths may be changed without altering the sizes of heat transfer surfaces. For instance, in the case of a plate heat exchanger, the dimensionless length can be changed by connecting the flow channels differently, i.e. without affecting either plate sizes or distances between plates. Also with the spiral-type heat exchanger the number of parallel flows may be changed by external modifications, not affecting the local geometry between tubes within the apparatus.

If the dimensionless length of a heat exchanger is increased at constant heat transfer area, the flow area for water diminishes, so that the flow velocity increases and the heat transfer improves. At the same time the pressure drop of the fluid rises, because of the greater flow velocity, and because of the greater flow path itself. This consideration suggests that for a given heat exchanger there must be some 'best' choice of heat exchanger length. Below we shall examine how lengths L_1^* and L_2^* should be selected to maximize the net electric output P_{e2}^* from our system.

If the dimensionless length and the dimensionless heat exchanger surface area for the consumer's heat exchanger are both changed at reference DH mass flowrate ($m^* = 1$), then water velocities over the heat transfer surface will vary according to:

$$v_{o,2}/v_{or,2} = L_2^*/S_2^* \quad (4-27)$$

Pressure drops per unit length of flow will consequently vary in proportion to L_2^{*2}/S_2^{*2} , so that for the total pressure drop over the heat exchanger we may write:

$$\frac{\Delta p_{o,2}}{\Delta p_{or,2}} = L_2^{*3}/S_2^{*2} \quad (4-28)$$

which means that the part $z_{ec,o}^*$ (changed dimensions and $m^* = 1$) of the total pumping power dissipated in the consumer's heat exchanger can be calculated from the expression:

$$z_{ec,o}^* = z_r L_2^*{}^3 S_2^{*-2} P_{ec,or}^* \quad (4-29)$$

giving:

$$z = z_r L_2^*{}^3 S_2^{*-2} P_{ec,or}^* / P_{ec,o}^* \quad (4-30)$$

and analogously:

$$w = w_r L_1^*{}^3 S_1^{*-2} P_{ec,or}^* / P_{ec,o}^* \quad (4-31)$$

The last two equations may be combined with eqs. (4-25) and (4-26) to give:

$$P_{ec,o}^* = P_{ec,or}^* (w_r L_1^*{}^3 S_1^{*-2} + z_r L_2^*{}^3 S_2^{*-2} + (x_r + y_r) L^* D^{*-5}) \quad (4-32)$$

When calculating how changes in heat exchanger sizes affect system elements outside the DH system proper, we shall as before simplify by neglecting both steam-side pressure drops in the turbine condenser and the pressure drops on the secondary side of the consumer's heat exchanger.

As when we calculated load performance variations in section 4.1, we shall further neglect the relatively small influence of variations in medium temperature on the heat transfer rates. Finally, we shall assume that when heat transfer rates on one side of a heat exchanger are altered, because of changed values of length and surface area, the overall heat transfer rates vary in the same proportion. This last assumption requires that changes in flow areas of one side of a heat transfer surface are accompanied by corresponding changes in flow areas on the other side. This specialization is made to avoid further complications of an already rather involved analysis, but it is clear that independent variations of flow areas on the two sides of the heat exchangers would be of practical interest. Traupel [4-5] has investigated this type of heat exchanger variation,

as applied to e.g. the steam generator of a gas-cooled reactor power plant.

Since the heat transfer coefficient on the primary side of the consumer's heat exchanger varies with flow velocity according to:

$$U_{2A,o}/U_{2A,or} = (v_o/v_{or})^{p_2} \quad (4-33)$$

we get, assuming that the overall heat transfer coefficient varies equally:

$$U_{2,o}/U_{2,or} = L_2^{*p_2} S_2^{*-p_2} \quad (4-34)$$

The dimensionless overall heat transfer coefficient is defined as:

$$U_{2,or}^* = \frac{U_{2,or} S_{2,r}}{mc} \quad (4-35)$$

so that at altered system dimensions it takes the value:

$$U_{2,o}^* = U_{2,or}^* L_2^{*p_2} S_2^{*1-p_2} \quad (4-36)$$

and analogously:

$$U_{1,o}^* = U_{1,or}^* L_1^{*p_1} S_1^{*1-p_1} \quad (4-37)$$

Presentation of results

Fig. 4-15 shows the effects of varying each of the three heat loss components (defined in section 3.2) on net electric output and on DH total heat loss. The three pairs of diagrams are the results of varying in turn, each of the three dimensionless insulation numbers I_{IO}^* , I_{IIO}^* , and I_{III}^* in table 4-4.

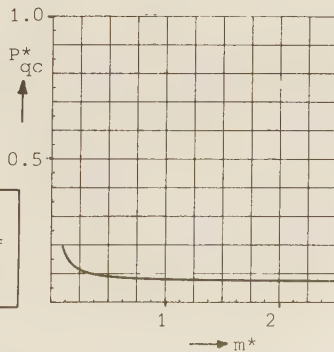
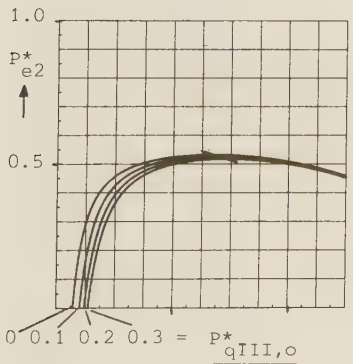
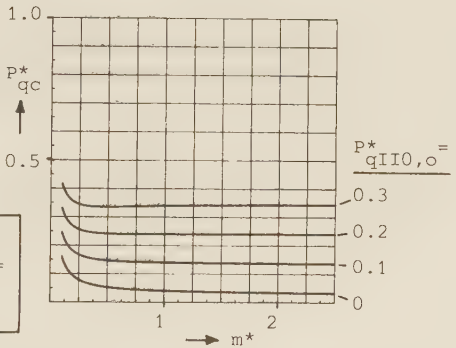
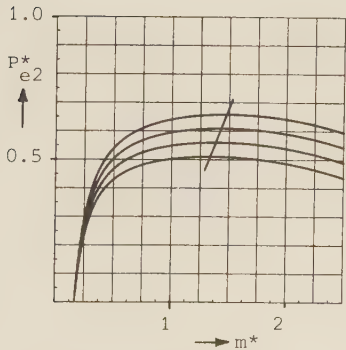
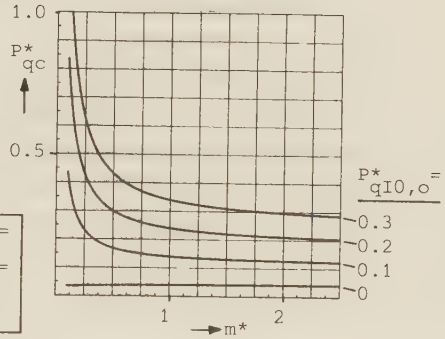
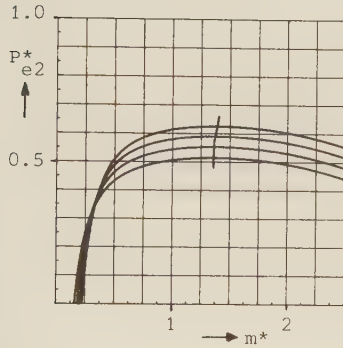
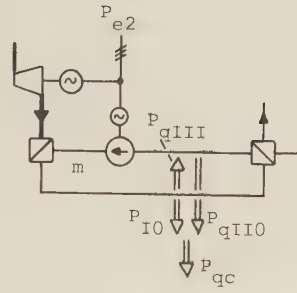
Fig. 4-15

Variation of DH heat loss components in simple system model. $t_a = 0^\circ\text{C}$.

P_{qI0} = heat loss from supply pipe

P_{qII0} = heat loss from return pipe

P_{qIII} = internal heat loss from supply to return pipe



From the figure we can establish that increases of the heat loss $P_{q_{IO,o}}^*$ from the supply and of the heat loss $P_{q_{IIO,o}}^*$ from the return line both have a positive effect on the net electric output: For a given heat demand $P_{q_2}^*$ at the consumer(s), the heat demand $P_{q_1}^*$ at the turbine condenser becomes greater, so that more steam can pass the turbine. We also find that the greatest increase of P_{e2}^* is achieved by allowing for a greater heat loss from the return line. The explanation is that when increasing $P_{q_{IO,o}}^*$, the positive influence upon $P_{q_1}^*$ is partly counteracted by a higher supply temperature t_{out1} at the turbine condenser, due to a greater temperature drop from t_{out1} to t_{out2} in the supply line.

This comparison illustrates the importance of distinguishing, as we have done in our thermodynamic model, between heat losses from supply and return lines, when DH heat generation is provided by a back-pressure plant. Often only total heat losses are considered in literature. The results also demonstrate that it is important to calculate, as our model equations include, the changes in supply and return temperature along the pipelines caused by energy exchanges for the fluid.

The practical conclusion from the comparison between the top and centre pair of diagrams of fig. 4-15 is that when selecting insulation thicknesses for supply and return lines from a back-pressure plant, a high rating of electric power gives an incentive to favour thermal insulation of the supply line. Even when heat generation is served by a heat-only boiler, such an incentive will exist because of the greater difference in temperature between supply water and the soil, as compared with that between return water and soil.

The fact that both increases of $P_{q_{IO,o}}^*$ and $P_{q_{IIO,o}}^*$ have a positive influence upon P_{e2}^* shows that when a DH system is served by a back-pressure turbine, the heat loss P_{qc}^* from the DH pipelines cannot be regarded as a pure loss: It may be the basis for an extra generation of electric power. Naturally, this extra power is generated with a low marginal efficiency for electricity, as compared with electricity generated in a condensing plant where heat is removed from the cycle at a lower temperature than in the case of a back-pressure plant.

At the bottom of fig. 4-15 are shown the effects of varying the third of the heat loss components, $P_{q_{III,o}}^*$, as would for instance be of interest when at the design stage considering variation of the distance between supply and return pipes in a common insulation. A higher value of $P_{q_{III,o}}^*$ is seen to lead to a deterioration of P_{e2}^* . Thus, although the internal heat loss component P_{III}^* does not affect the total heat loss P_{qc}^* from the pipeline, it is not unimportant. In practice the negative effects from allowing a greater internal heat loss $P_{q_{III}}^*$ must be weighed against the benefit in the form of lower investment costs for pipelines that may follow from smaller external conduit dimensions.

Figs. 4-16 and 4-17 show effects of varying pipeline length and diameter on net electric output and on DH heat losses. In the first of these two figures, the DH mass flowrate m^* has been selected for the abscissa, in the second the supply temperature is the abscissa instead. As before the two types of representation complement each other to give a comprehensive picture of system characteristics.

Three cases are analysed in the two figures:

- Combined variation of system length L^* and diameter D^* assuming that the system is designed for vanishing heat loss components.
- Variation of L^* only with non-vanishing values of heat loss components (in accordance with the values specified in table 4-5).
- Variation of D^* only with non-vanishing values of heat loss components.

In the case of vanishing heat loss components, separate diagrams could have been derived for variations of L^* and D^* . However, by choosing L^*/D^{*5} for curve parameter a more condensed representation is achieved. When heat loss components do not vanish, two sets of diagrams cannot be avoided.

From the diagrams it can be seen that in all cases where the flow resistance of the DH system is increased, i.e. when L^* is increased and when D^* is

Fig. 4-16

Variation of DH dimensionless length L^* and of dimensionless diameter D^* . $t_a = 0^\circ\text{C}$.

$$D^* = \frac{D_x}{D_y} = \frac{D_{x,r}}{D_{y,r}}$$

$$= \frac{D_x}{D_y} = \frac{D_{x,r}}{D_{y,r}}$$

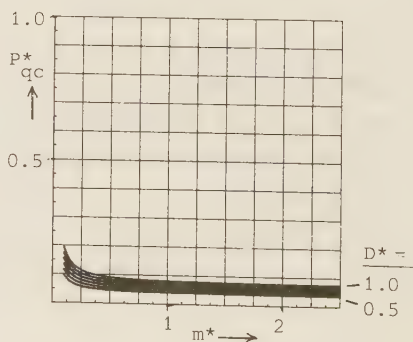
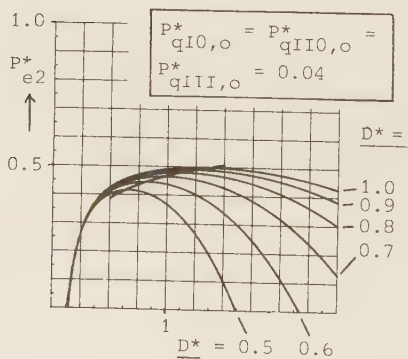
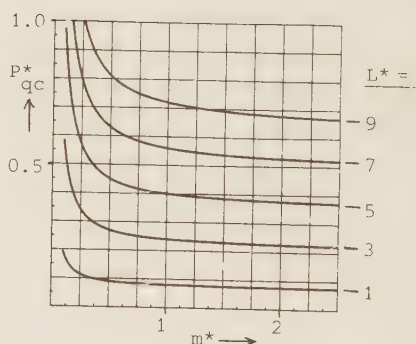
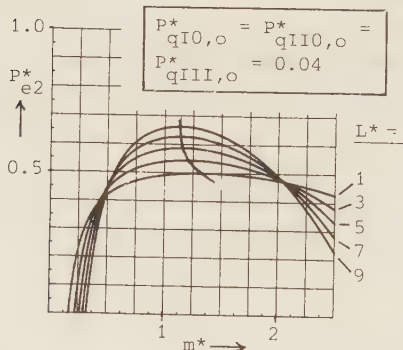
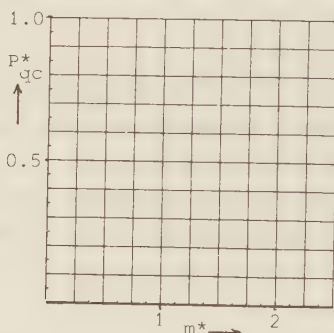
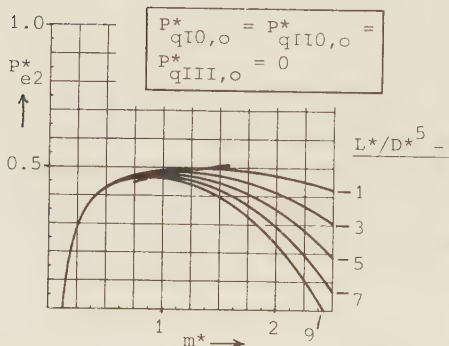
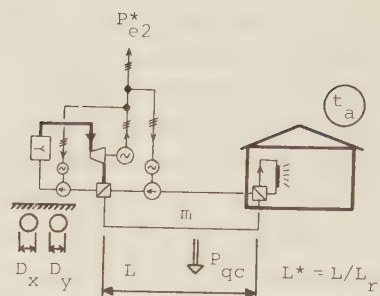
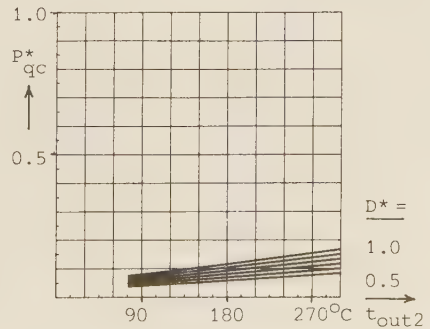
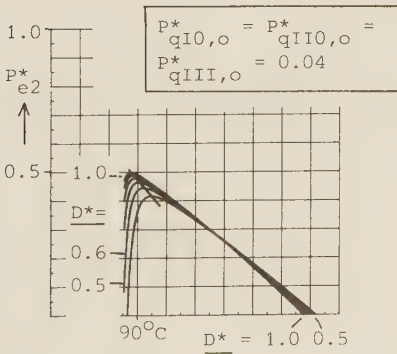
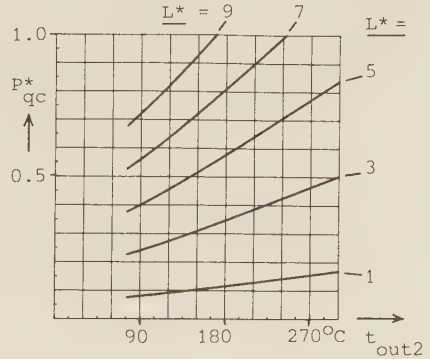
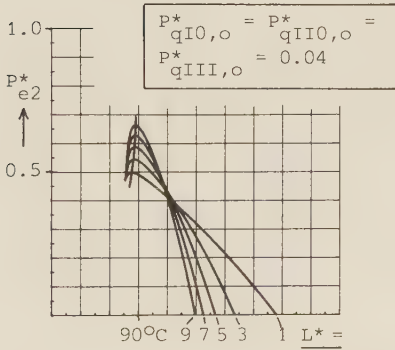
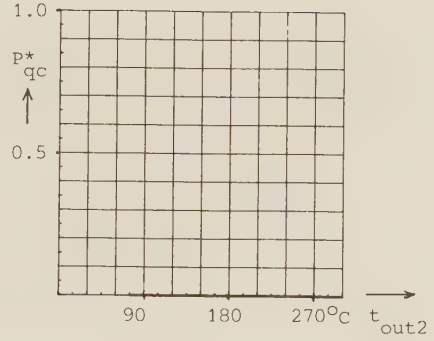
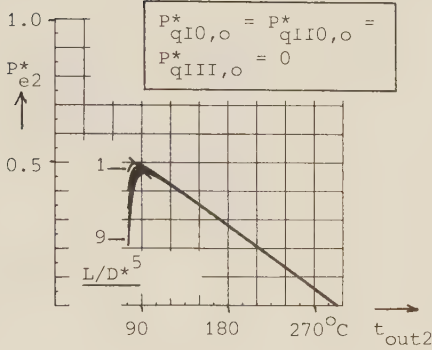
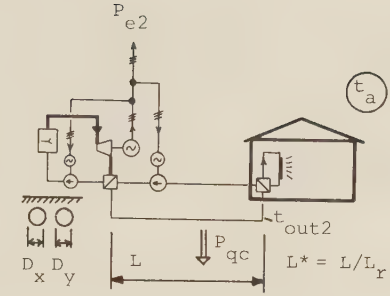


Fig. 4-17

Variation of DH dimensionless length L^* and of dimensionless diameter D^* . $t_a = 0^\circ\text{C}$.

$$D^* = D_x^* = D_x/D_{x,r}$$

$$= D_y^* = D_y/D_{y,r}$$



diminished, maximum net electric output P_{e2}^* requires a smaller DH flowrate m^* and a correspondingly higher supply temperature t_{out2} . This variation of optimal m^* is seen to be in accordance with the approximate equation (4-17), derived in section 4.1. Here, a greater L^* or a smaller D^* correspond to a greater value of the pumping loss $P_{ec,o}^*$ at reference DH mass flowrate.

Total heat losses P_{qc}^* are seen to vary in figs. 4-16 and 4-17 as would be expected and are shown to complete the picture.

We have found that a pure thermodynamic performance optimization criterion (maximum P_{e2}^*) leads to higher optimal supply temperatures at longer transmission distances for hot water. The same tendency is found when deciding on the supply temperature on the grounds of economic considerations, including investment costs for pipelines [3-8].

The analysis given here of changing pipe length and diameter in the case of a single pair of supply and return pipes can be interpreted as representative for some cases of variation of corresponding dimensions of pipes in branched systems, while other such cases are clearly beyond the simple model (cf. section 2.4). If for instance at the design stage all pipe diameters of a branched system are considered to be varied by the same factor, variation of D^* in our model will be a reasonable representation of the empirical system. As an example where our simple model will not be immediately applicable, one may consider the case of system size increasing due to a growing number of buildings being connected to the system. Here, heat and pumping losses will in general be affected differently from what the system of equations in table 4-4 assumes. For instance, adding pipes of small diameter will increase heat losses relatively more than it will increase pumping losses.

The question of how to select the optimal diameter of pipes in which a fluid is transported, represents a classical optimization problem, occurring in various branches of engineering. Pipelines for cold-water supply in built-up areas would be one such example. Fig. 4-18 shows qualitatively how an economic optimum may be found for the diameter of a DH pipe from a heat-only boiler. Here, a cost function is defined as the sum of costs for pipe-

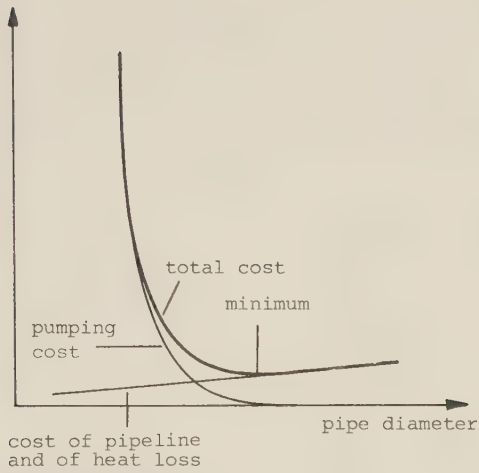


Fig. 4-18 Economic pipe diameter optimization. Classical procedure where mass flowrate is (often tacitly) kept constant.

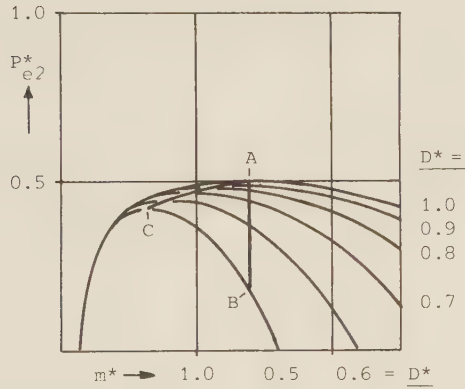


Fig. 4-19 Change of net electric output P_{e2} with pipe diameter.
 A-B: constant mass flowrate m
 A-C: mass flowrate m adjusted to maximize P_{e2} .

line installation, for heat losses, and for pumping power. Of these three cost elements, the two first increase their values moderately with diameter, while the pumping power drops rapidly with increasing diameter.

Optimizations of this kind usually assume that the flowrate of the fluid is kept constant when the diameter is varied. While this assumption can be argued as reasonable in the case of heat generation in a heat-only boiler, it would be misleading, when the heat is generated in a back-pressure plant. As is illustrated by fig. 4-19, keeping the flowrate m^* constant when diminishing the diameter will reduce the net electric output unnecessarily. By partly compensating for the greater network flow resistance by diminishing m^* , a much smaller reduction of P_{e2}^* is achieved.

Results from varied heat exchanger dimensions in our model are shown in diagrams 4-20, 21, 22, and 23. Here the technique of utilizing contour curves has been used extensively. In this way it has been possible to condense a fairly large amount of numerical results on a limited space. A similar kind of graphical representation was adopted by Knizia [4-6], when he investigated the question of selecting intermediate steam pressures for a steam cycle with double reheating; here, the two pressures were used for diagram axes, and contour curves were derived for absolute and specific plant steam consumption.

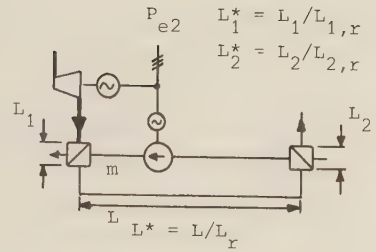
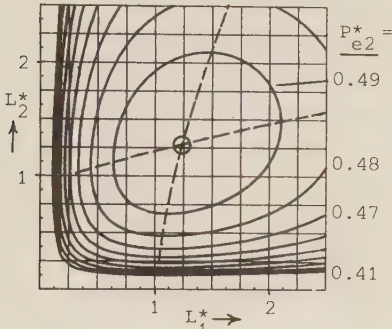
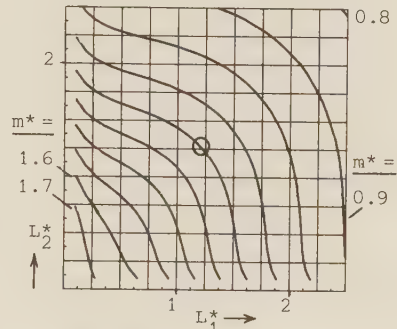
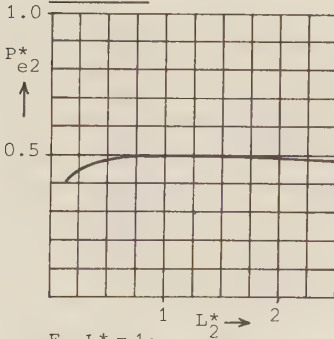
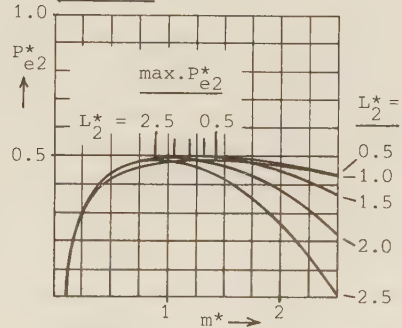
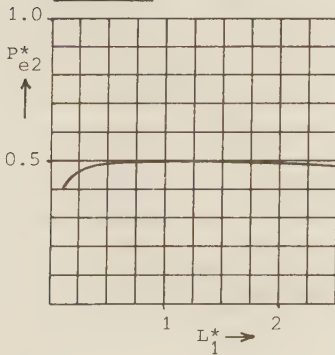
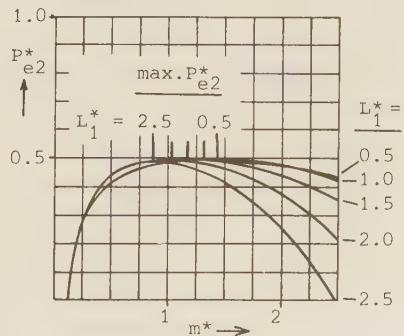
In the upper left-hand corner of fig. 4-20 (diagram A) are shown contour curves for the net electric output P_{e2}^* with the dimensionless heat exchanger lengths L_1^* and L_2^* as abscissa and ordinata, respectively. Unlike earlier representations of system characteristics, the DH water flowrate here no longer varies freely: For each combination of values of L_1^* and L_2^* , P_{e2}^* is given for that value of m^* which maximizes P_{e2}^* . Thus, m^* here becomes a variable which is dependent upon L_1^* and L_2^* . This functional relationship is given in the upper right-hand corner in fig. 4-20 (diagram B), where contour curves are shown for m^* . In diagram A dotted curves, connecting points of horizontal and vertical tangents for P_{e2}^* contour curves, have been plotted.

Diagrams C and E of fig. 4-20 represent vertical and horizontal 'cuts' in diagram A above, specializing on the cases of $L_1^* = 1$ and $L_2^* = 1$, respectively.

Fig. 4-20

Variation of dimensionless heat exchanger lengths L_1^* and L_2^* and of dimensionless DH water flowrate m^* to maximize net electric output P_{e2}^* .

$t_a = 0^\circ\text{C}$, $L_1^* = S_1^* = S_2^* = 1$.

A. m^* varied to maximize P_{e2}^* :B. m^* varied to maximize P_{e2}^* :C. $L_1^* = 1$:D. $L_1^* = 1$:E. $L_2^* = 1$:F. $L_2^* = 1$:

In diagram D and F, it is shown how P_{e2}^* varies with DH flowrate m^* for selected values of heat exchanger lengths L_1^* and L_2^* .

From diagram A the following interpretation can be made: There exists an optimum set of values for heat exchanger lengths L_1^* and L_2^* in the sense of maximum P_{e2}^* . The contour curves are almost symmetrical around a line of 45° slope, so that variations in L_1^* and L_2^* will affect P_{e2}^* rather equally. The absolute maximum for P_{e2}^* lies at values of L_1^* and L_2^* slightly above 1.0, so that in the numerical example of the figure it will be optimal to let a greater part of the total pumping loss P_{ec}^* be dissipated in the heat exchangers, as compared with losses in the pipelines. (At reference system dimensions, i.e. when $L_1^* = L_2^* = 1$, x , y , z , and w were all assigned the value of 0.25, as specified in table 4-5.) In DH it is normal to design heat exchangers for primary side pressure losses of only a fraction of pressure drops in pipelines. Due to the existence of control valves in branched systems (cf. section 2.4), our result for optimum L_1^* and L_2^* cannot be transferred directly to such systems. Still, our result may suggest that practices regarding choice of heat exchanger pressure drops be reexamined.

The curves of diagrams D and F of fig. 4-20 can be compared with the previous findings for P_{e2}^* in fig. 4-16, where not heat exchanger lengths, but DH pipe length L^* was varied. We see that when heat exchanger lengths are increased, so that the total flow resistance of the DH system goes up, maximum P_{e2}^* requires that m^* be diminished, as when L^* was increased. Comparing the top diagram in fig. 4-16 with diagrams D and F of fig. 4-20 we also notice an interesting difference in the variation of maximum P_{e2}^* : Whereas maximum P_{e2}^* steadily goes down with increased length L^* , it has a maximum when increasing instead L_1^* or L_2^* . The explanation for this difference in variation is that in the case of increased heat exchanger lengths, the negative influence of greater flow resistance upon P_{e2}^* is counteracted by an improvement of heat transfer, whereas increasing pipe length only increases flow resistance.

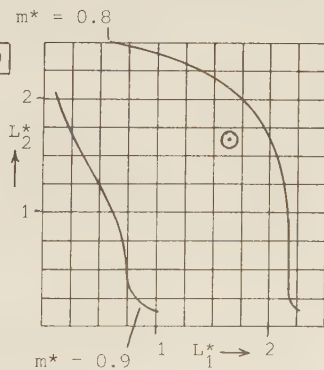
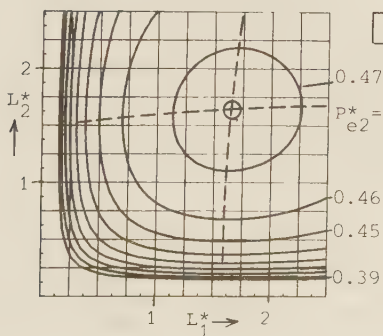
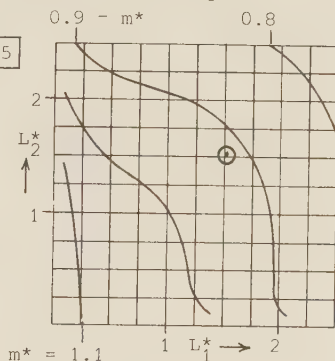
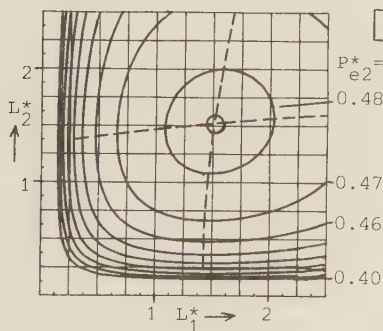
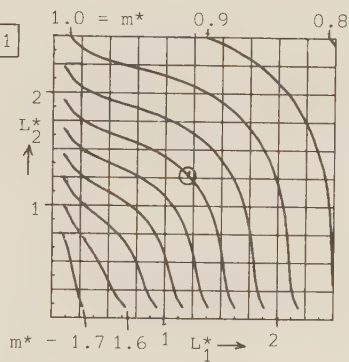
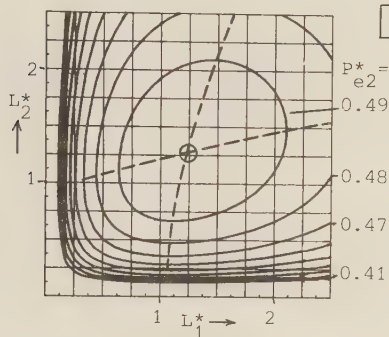
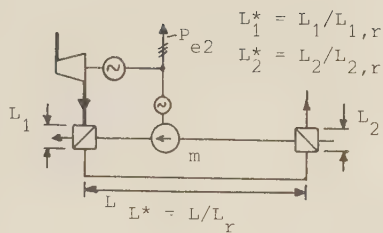
In practical systems acceptable pressure drops over heat exchangers will be limited by erosion, noise, etc. It will be important to observe, however, that it is not so much the total pressure drop over the apparatus that is critical in this respect, but rather the pressure drop per unit flow length, or possibly the amount of mechanical power dissipated per unit vol-

Fig. 4-21

Varying heat exchanger lengths L_1^* , L_2^* , and varying pipeline length L^* .

m^* changing with L_1^* , L_2^* , L^* to maximize P_{e2}^* .

$t_a = 0^\circ\text{C}$, $S_1^* = S_2^* - 1$.



ume. The total pressure drop over a long heat exchanger may be great, and still the pressure drop per unit length of flow may be moderate. When in diagram A in fig. 4-20 L_1^* or L_2^* is increased, m^* (as seen from diagram B) decreases, so that variations in pressure drops become rather moderate.

The results shown in fig. 4-20 were derived for an ambient temperature t_a of 0°C . As we have seen in section 4.1, a lowering of t_a leads to an increased value of the mass flowrate m^* to maximize P_{e2}^* . Hereby pressure drops over heat exchangers also increase. Thus, if a limit value is prescribed for the pressure drop per unit length of flow in the heat exchangers, this may restrict the choice of m^* at low ambient temperatures, a fact which must be taken into consideration when deciding on lengths L_1^* and L_2^* of heat exchangers.

In fig. 4-21 the contour plots for P_{e2}^* and m^* of fig. 4-20 have been repeated for various values of the pipeline length L^* . It can be seen that the absolute maximum for P_{e2}^* (where the dotted lines intersect) moves towards higher values of L_1^* and L_2^* , when L^* is increased. That is, when the flow resistance of the DH pipelines is increased, flow resistances in the heat exchangers should follow suit to a certain extent. Still, when L^* becomes larger, a relatively greater part of the total pumping power P_{ec}^* should be dissipated in the pipelines, i.e. fraction numbers x and y are increased.

The fact that optimal values for L_1^* and L_2^* increase with increasing L^* can be explained as follows: At each value of L^* the choices of L_1^* and L_2^* must be made from a weighting of two counteracting tendencies against each other: Increasing heat exchanger lengths cause greater flow resistance in the apparatuses (tending to increase P_{ec}^*), but at the same time the heat transfer improves, due to greater flow velocity of the fluid over the heat transfer surface. Now, when L^* is increased, the benefit from the latter of these two opposing effects becomes more important, since improved heat transfer in heat exchangers means that the DH mass flowrate can be adjusted downwards, leading to a smaller dissipated pumping power in the pipelines, a fact that becomes increasingly important when the pipelines are long.

Two further observations can be made from fig. 4-21: The first is that with increasing L^* , the value of m^* to maximize P_{e2}^* goes down, which is in ac-

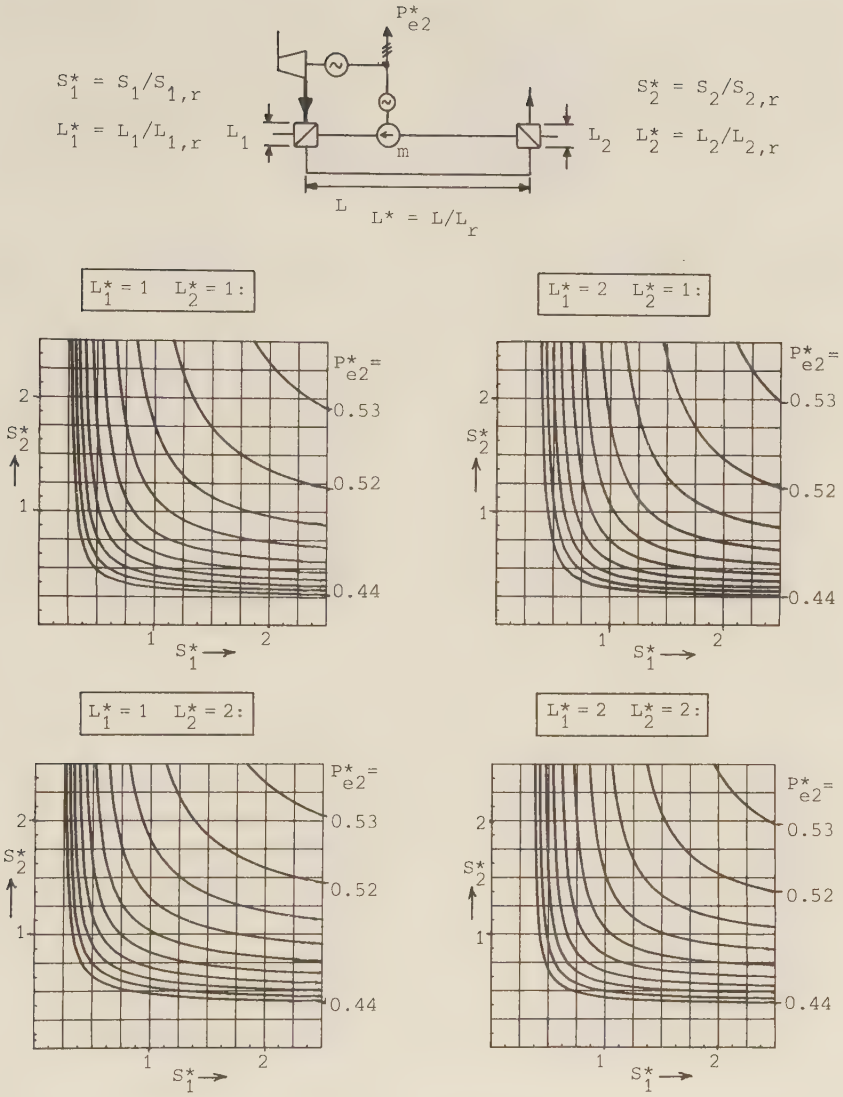


Fig. 4-22

Simultaneous variation of dimensionless heat exchanger surface areas and dimensionless heat exchanger lengths.

m^* changing with S_1^* , S_2^* , L_1^* , and L_2^* to maximize P_{e2}^* .

$t_a = 0^\circ\text{C}$ $L^* = 1$.

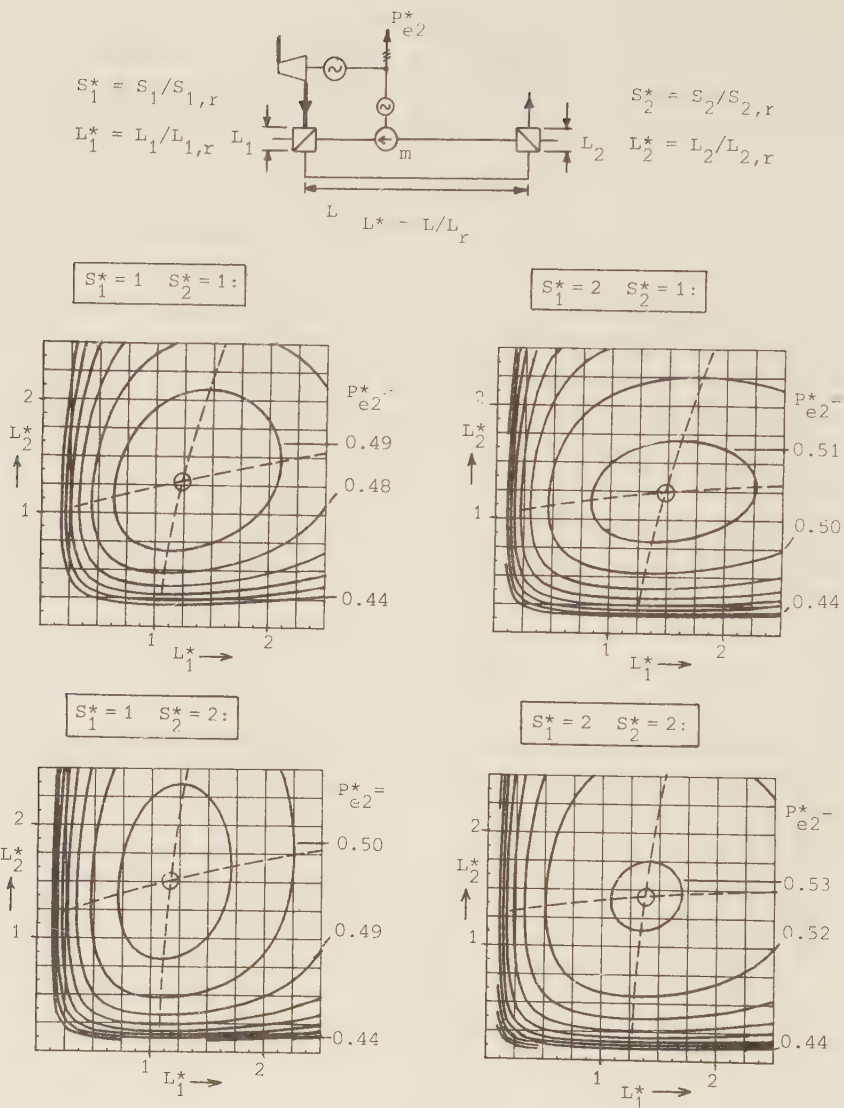


Fig. 4-23

Simultaneous variation of dimensionless heat exchanger surface areas and dimensionless heat exchanger lengths.

m^* changing with S_1^* , S_2^* , L_1^* , and L_2^* to maximize P_{e2}^* .

$t_a = 0^\circ\text{C}$ $L^* = 1$.

cordance with previous findings in fig. 4-16. Secondly, when L^* increases in fig. 4-21, the distances between the contour curves for m^* become greater. The reason is that with increasing L^* , changes in L_1^* and L_2^* become less important for the total flow resistance of the DH system, so that the necessary adjustments to maximize P_{e2}^* become smaller.

In the figures shown so far, where consequences of altering dimensions of heat exchangers have been shown, the dimensionless heat transfer surface areas S_1^* and S_2^* were kept constant. In figs. 4-22 and 4-23 are shown consequences of varying all the four heat exchanger dimensions L_1^* , L_2^* , S_1^* , and S_2^* simultaneously. As before, contour curves for P_{e2}^* are given, specializing the variation of m^* to those values which at each combination of system dimensions maximize P_{e2}^* .

The contour curves of fig. 4-22 are open curves, i.e. P_{e2}^* improves steadily with increasing values of heat exchanger surface size. The marginal gains, however, level out with increasing values of S_1^* and S_2^* .

In fig. 4-23 the contour curves are closed. By following how the absolute maximum for P_{e2}^* shifts between each of the four diagrams, we see that obviously increases in heat transfer area should be followed by increases in heat exchanger length. The contour curves are seen to be almost symmetrical around a line of 45° slope, so long as $S_1^* = S_2^*$, whereas for $S_1^* \neq S_2^*$ more ellipse-like curves are found. It can be seen that the smallest heat exchanger (which is more sensitive to changes in heat transfer rate) is more critical with respect to choice of length, that is P_{e2}^* changes more with heat exchanger length.

5. LOAD PERFORMANCE OF EXTENDED MODEL

5.1 General

This chapter is devoted to a load performance analysis of the extended system model, presented in Chapter 2 and shown again in fig. 5-1. By virtue of including consumption hot water services and a peaking heat-only boiler, the extended model is much closer to practical systems than is the simplest model.

As an alternative to a heat-only boiler, peaks in heating demand can be covered by utilizing throttled live steam, as shown in fig. 5-2. In that case the plant boiler must be designed for a greater maximum steam output. With some minor deviations the results arrived at for the extended model will apply to this variant as well. In principle, throttling of steam will lead to a smaller electrical output at great heat loads, due to a greater power consumption of the boiler feed pump. Also the two types of solution may have somewhat different total fuel consumption rates, depending on, for instance, if the heat-only boiler is equipped with an air preheater or not. However, we shall be concerned in this chapter primarily with the electrical output from the plant.

The fact that a back-pressure plant is nearly always equipped with some kind of peaking heat production facility, derives from the form of duration curves for heating of buildings, fig. 5-3. It can be seen that loads above 50% of maximum occur only rarely, so that if the back-pressure plant is designed for no more than this load, it may still be capable of providing more than 90% of the total heat energy of a heating season. Heat load duration curves usually display the greatest variation of slope around 50% load, so that it is very common [5-1], [5-2] to use this fraction as a design target for the back-pressure plant. The optimum fraction from an economical point of view depends on a number of factors, among them the ratio between specific investment costs for back-pressure production and for peaking heat production capacity.

Despite this type of design target, back-pressure plants may sometimes in practice cover a substantially greater or smaller part of maximum heat load. The reason is that DH networks are usually expanded over many years, so that if only a single back-pressure is installed, it may initially be quite large,

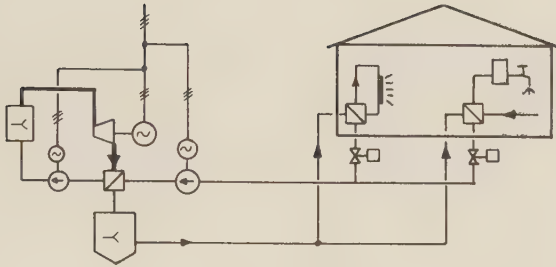


Fig. 5-1 Extended system model, presented in section 2.1.

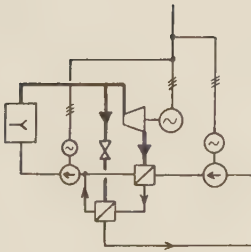


Fig. 5-2

Back-pressure plant with peaking heat generation capacity by throttling of live steam.

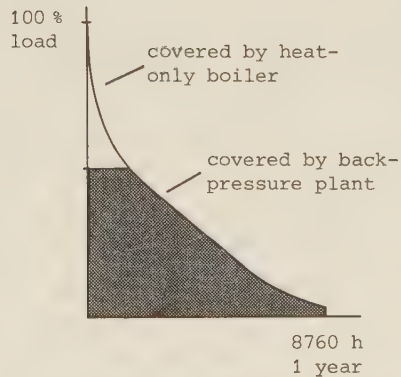


Fig. 5-3

Typical load duration curve for heat to a DH network. Peak load covered by heat-only boiler.

compared to the size of the heat load. When in subsequent years the size of the heat load grows, the relative size of the turbine diminishes accordingly.

An alternative would be to split up the back-pressure plant into two or more units and install them gradually, to obtain a better matching between the size of the heat load and the installed capacity for back-pressure production. In this way poor utilization of part of the installed capacity can be avoided, but at the same time the specific investment cost for smaller plants tends to be larger. Thus it cannot be told straight away which number of plants is optimal. One town, which has driven the principle of dividing up CHP capacity quite far, is Flensburg in the FRG [5-3]. Provided several turbines are connected in parallel to the DH network, our model calculations may still be considered valid. However, it leads to a higher electrical output to adopt series connection, as has been done in Flensburg and in most other places where more than one turbine has been installed.

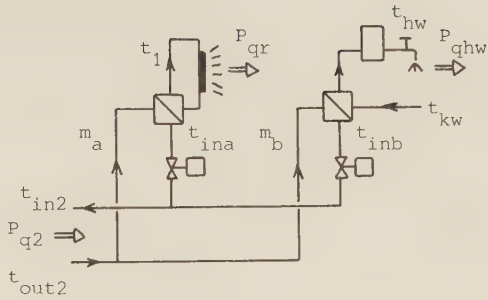
As can be seen from fig. 5-1, the allotment of partial DH water flows for the two heat exchangers in the building is supposed to be controlled by primary valves in series with the heat exchangers. To simplify our calculations, we shall assume that pressure drops in both control valves and in heat exchangers are so small that it is justified to neglect them. Consequently, the DH pumping power P_{ec} will be needed only to cover pressure losses in the supply and return pipes. In spite of this simplification we shall allow the operational mode of the DH to be varied freely, as when we analysed load performance of the simple system model in section 4.1. The considerations for technological constraints developed in section 4.2 will naturally apply to our extended model as well. In a more accurate analysis, effects of great pressure drops in heat exchangers (as discussed in section 3.3) and in primary control valves would have to be implemented.

The analysis in the following will specialize in cases where the demand for hot consumption water remains constant. As was pointed out already when the extended model was presented in Chapter 2, it is possible to reduce the impact of intermittent hot water consumption on the DH network, by employing a storage tank in the building. Nevertheless it would be of great practical interest to investigate consequences of variations in hot water consumption.

5.2 Derivation of Equations

Fig. 5-4

Consumer's connection of extended system model.



In the case of the extended model the total heat load P_{q2} from the building(s) results from the addition of two components:

$$P_{q2} = P_{qr} + P_{qhw} \quad (5-1)$$

from the radiator heat exchanger and the heat exchanger for hot water provision, respectively. The total primary mass flowrate m at the consumer's connection branches into two flows:

$$m = m_a + m_b \quad (5-2)$$

Energy flows are made dimensionless by dividing them by the total heat load $P_{q2,o}$ at reference load conditions (shown by index o in consistence with earlier nomenclature):

$$P_{q2}^* = P_{q2} / P_{q2,o} \quad (5-3)$$

$$P_{qr}^* = P_{qr} / P_{q2,o} \quad (5-4)$$

$$P_{qhw}^* = P_{qhw} / P_{q2,o} \quad (5-5)$$

From these equations we may write, assuming that $P_{qhw} \equiv P_{qhw,o}$:

$$P_{q2}^* = P_{qr}^* + 1 - P_{qr,o}^* \quad (5-6)$$

Dimensionless DH mass flowrates are introduced according to:

$$m^* = m/m_o \text{ (as with the simplest model)} \quad (5-7)$$

$$m_a^* = m_a/m_{a,o} \quad (5-8)$$

$$m_b^* = m_b/m_{b,o} \quad (5-9)$$

By defining the mass flow ratio:

$$M_{ab} = \frac{m_{a,o}}{m_{a,o} + m_{b,o}} \quad (5-10)$$

for which we may write:

$$M_{ab} = \left(1 + \frac{1 - p_{gr,o}^*}{p_{gr,o}^*} \frac{t_{out2,o} - t_{ina,o}}{t_{out2,o} - t_{inb,o}}\right)^{-1} \quad (5-11)$$

we may express the dimensionless total DH mass flowrate in terms of the dimensionless mass flowrates to the radiator and hot water heat exchangers:

$$m^* = M_{ab} m_a^* + (1 - M_{ab}) m_b^* \quad (5-12)$$

The DH return water temperature, t_{in2} , results from mixing the return flows from the two heat exchangers:

$$t_{in2} = \frac{m_a}{m_a + m_b} t_{ina} + \frac{m_b}{m_a + m_b} t_{inb} \quad (5-13)$$

or, in terms of dimensionless mass flowrates:

$$t_{in2} = \frac{M_{ab} m_a^*}{m^*} t_{ina} + \left(1 - \frac{M_{ab} m_a^*}{m^*}\right) t_{inb} \quad (5-14)$$

In the case of the simplest system model the dimensionless heat transfer coefficient, $U_{2,o}^*$, at reference load conditions was assigned a numerical value at the calculation procedure, and the corresponding DH water return temperature $t_{in2,o}$ was calculated. In the present case it will instead be convenient to specify numerically the water temperatures $t_{out2,o}$, $t_{ina,o}$, and $t_{inb,o}$ and from those temperatures calculate the two dimensionless heat transfer coefficients $U_{r,o}^*$ and $U_{hw,o}^*$ at reference load. From the energy balance equation:

$$P_{qr,o} = m_{a,o} c_p (t_{out2,o} - t_{ina,o}) = \frac{t_{out2,o} - t_{ina,o}}{\ln \frac{t_{out2,o} - t_{1,o}}{t_{ina,o} - t_{1,o}}} U_{r,o} A_r \quad (5-15)$$

$$\text{we get: } U_{r,o}^* = \frac{U_{r,o} A_r}{m_{a,o} c_p} = \ln \frac{t_{out2,o} - t_{1,o}}{t_{ina,o} - t_{1,o}} \quad (5-16)$$

Analogously the energy balance:

$$\begin{aligned} P_{qhw,o} &= m_{b,o} c_p (t_{out2,o} - t_{inb,o}) = \\ &= \frac{(t_{out2,o} - t_{hw}) - (t_{inb,o} - t_{kw})}{\ln \frac{t_{out2,o} - t_{hw}}{t_{inb,o} - t_{kw}}} U_{hw,o} A_{hw} \end{aligned} \quad (5-17)$$

for the hot water exchanger yields the equation:

$$U_{hw,o}^* = \frac{U_{hw,o} A_{hw}}{m_{b,o} c_p} = \frac{t_{out2,o} - t_{inb,o}}{(t_{out2,o} - t_{hw}) - (t_{inb,o} - t_{kw})} \ln \frac{t_{out2,o} - t_{hw}}{t_{inb,o} - t_{kw}} \quad (5-18)$$

In analogy to the expression derived for the dimensionless heat transfer coefficient U_2^* in the simple model without hot water production, we may write:

$$U_r^* = \frac{U_{r,o}^*}{m_a^*} \frac{1 + x_r}{m_a^* - p_r + x_r} \quad (5-19)$$

and:

$$U_{hw}^* = \frac{U_{hw,o}^*}{m_b^*} \frac{1 + x_{hw}}{m_b^{*-p_{hw}} + x_{hw}} \quad (5-20)$$

where x_r and x_{hw} are ratios between heat transfer coefficients of the two sides of the respective heat transfer surfaces, and p_r and p_{hw} are exponents in equations describing variations in heat transfer coefficients with flow velocities.

Utilizing the equation:

$$t_{inb} - t_{out2} = (t_{inb,o} - t_{out2,o}) / m_b^* \quad (5-21)$$

and:

$$\frac{t_{inb} - t_{out2} + t_{hw} - t_{kw}}{t_{out2} - t_{hw}} = \exp \left[\left(\frac{t_{hw} - t_{kw}}{t_{out2} - t_{inb}} - 1 \right) U_{hw}^* \right] - 1 \quad (5-22)$$

we may write down the following equation:

$$\frac{(t_{inb,o} - t_{out2,o}) / m_b^* + t_{hw} - t_{kw}}{t_{out2} - t_{hw}} = \exp \left[\left(m_b^* \frac{t_{hw} - t_{kw}}{t_{out2,o} - t_{inb,o}} - 1 \right) U_{hw}^* \right] - 1 \quad (5-23)$$

from which the dimensionless primary mass flowrate m_b^* to the hot water DH heat exchanger may be calculated iteratively.

So long as the heat supply P_{q1} from the back-pressure plant to the DH system is so small that the full steam flowrate, which we shall designate m_{st} , is not reached, the calculation of the cycle output will be identical to the procedure followed for the simplest steam cycle. But at some ambient temperature this is no longer so, and only part of the total heat supply, P_{qt} , is delivered by the steam cycle, see fig. 5-5. Now the steam condensing temperature is no longer determined primarily by the DH supply temperature t_{out1} , but by some intermediate temperature t_i between t_{out1} and the return temperature t_{in1} .

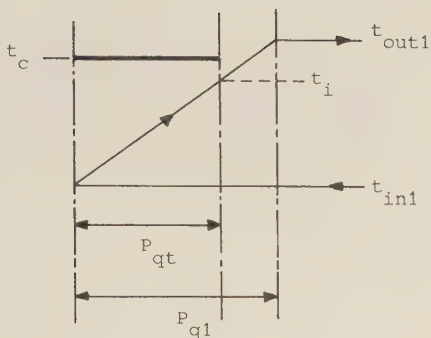


Fig. 5-5

Water heating in turbine condenser and in heat-only boiler. Temperatures plotted against transferred heat.

An expression for the condensing temperature t_c may be derived by combining the three equations:

$$P_{qt}^* = m_{st}^* P_{q1,o}^* (1 + b(t_c - t_{c,o})) \quad (5-24)$$

$$t_i - t_{in1} = (t_c - t_{in1}) (1 - \exp(-U_1^*)) \quad (5-25)$$

$$t_i - t_{in1} = (t_{out1} - t_{in1}) P_{qt}^* / P_{q1}^* \quad (5-26)$$

to the following equation:

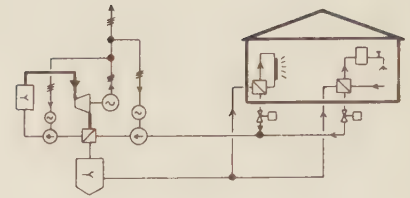
$$t_c = t_{c,o} + \frac{-(t_{c,o} - t_{in1}) (1 - \exp(-U_1^*)) + m_{st}^* \frac{P_{q1,o}^*}{P_{q1}^*} (t_{out1} - t_{in1})}{(1 - \exp(-U_1^*)) - b m_{st}^* \frac{P_{q1,o}^*}{P_{q1}^*} (t_{out1} - t_{in1})} \quad (5-27)$$

The intermediate water temperature t_i is given by the equation:

$$t_i = t_{in1} + (t_{out1} - t_{in1}) P_{qt}^* / P_{q1}^* \quad (5-28)$$

Table 5-1

System of equations describing load performance of extended system model.



$$P_{qr}^* = P_{qr,o}^* (1 + i(t_a - t_{a,o}))$$

$$t_1 = t_h + (t_{1,o} - t_h) (P_{qr}^* / P_{qr,o}^*)^{1/n}$$

$$U_{r,o}^* = \ln \frac{t_{out2,o} - t_{1,o}}{t_{ina,o} - t_{1,o}} \quad (5-16)$$

$$U_r^* = \frac{U_{r,o}^*}{m_a^*} \frac{1 + x_r}{\frac{-p_r}{m_a^*} + x_r}$$

$$t_{out2} = t_1 + (t_{out2,o} - t_{1,o}) \frac{P_{qr}^* / P_{qr,o}^*}{m_a^*} \frac{1 - \exp(-U_{r,o}^*)}{1 - \exp(-U_r^*)}$$

$$t_{ina} = t_{out2} - (t_{out2,o} - t_{ina,o}) \frac{P_{qr}^* / P_{qr,o}^*}{m_a^*}$$

$$U_{hw,o}^* = \frac{t_{out2,o} - t_{inb,o}}{(t_{out2,o} - t_{hw}) - (t_{inb,o} - t_{kw})} \ln \frac{t_{out2,o} - t_{hw}}{t_{inb,o} - t_{kw}} \quad (5-18)$$

$$U_{hw}^* = \frac{U_{hw,o}^*}{m_b^*} \frac{1 + x_{hw}}{\frac{-p_{hw}}{m_b^*} + x_{hw}} \quad (5-20)$$

$$\frac{\frac{1}{m_b^*} (t_{inb,o} - t_{out2,o}) + t_{hw} - t_{kw}}{t_{out2} - t_{hw}} =$$

$$= \exp \left[\left(\frac{t_{hw} - t_{kw}}{m_b^* t_{out2,o} - t_{inb,o}} - 1 \right) U_{hw}^* \right] - 1$$

From this equation m_b^* is solved iteratively

(5-23)

(to be continued)

Table 5-1, continued

$$t_{inb} = t_{out2} - (t_{out2,o} - t_{inb,o})/m_b^*$$

$$M_{ab} = (1 + \frac{1 - P_{qr,o}^*}{P_{qr,o}^*} \frac{t_{out2,o} - t_{ina,o}}{t_{out2,o} - t_{inb,o}})^{-1} \quad (5-11)$$

$$m^* = M_{ab} m_a^* + (1 - M_{ab}) m_b^* \quad (5-12)$$

$$t_{in2} = \frac{M_{ab} m_a^*}{m^*} t_{ina} + (1 - \frac{M_{ab} m_a^*}{m^*}) t_{inb} \quad (5-14)$$

$$t_{in2,o} = M_{ab} t_{ina,o} + (1 - M_{ab}) t_{inb,o}$$

$$P_{ec}^* = P_{ec,o}^* m^{*3}$$

$$j = P_{IO,o}^* / (t_{out2,o} - t_s)$$

$$P_{qIO}^* = P_{qIO,o}^* + j(t_{out2} - t_{out2,o})$$

$$k = P_{qIIO,o}^* / (t_{in2,o} - t_s)$$

$$P_{qIIO}^* = P_{qIII,o}^* + k(t_{in2} - t_{in2,o})$$

$$l_{10} = P_{qIII,o}^* / (t_{out2,o} - t_{in2,o})$$

$$l_{01} = -l_{10}$$

$$P_{qIII}^* = P_{qIII,o}^* + l_{10}(t_{out2} - t_{out2,o}) + l_{01}(t_{in2} - t_{in2,o})$$

$$P_{qc}^* = P_{qIO}^* + P_{qIIO}^*$$

$$P_{qc,o}^* = P_{qIO,o}^* + P_{qIIO,o}^*$$

$$P_{q2}^* = P_{qr}^* + 1 - P_{qr,o}^* \quad (5-6)$$

$$P_{q1}^* = P_{q2}^* + P_{qc}^* - P_{ec}^*$$

$$P_{q1,o}^* = 1 + P_{qc,o}^* - P_{ec,o}^*$$

$$t_{in1} = t_{in2} + (t_{out2} - t_{in2}) \frac{y P_{ec}^* - P_{qIIO}^* + P_{qIII}^*}{P_{q2}^*}$$

$$t_{in1,o} = t_{in2,o} + (t_{out2,o} - t_{in2,o}) (y P_{ec,o}^* - P_{qIIO,o}^* + P_{qIII,o}^*)$$

(to be continued)

Table 5-1, continued

$$t_{out1} = t_{in1} + (t_{out2} - t_{in2}) \frac{P_{q1}^*}{P_{q2}^*}$$

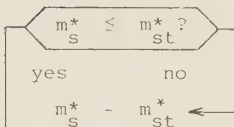
$$t_{out1,o} = t_{in1,o} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1,o}^*}{P_{q1,o}^*}$$

$$U_1^* = \frac{U_{1,o}^*}{m^*} \frac{1 + x_1}{-p_1 + x_1} \quad (5-19)$$

$$t_c = t_{in1} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1}^*}{m^*(1 - \exp(-U_1^*))}$$

$$t_{c,o} = t_{in1,o} + (t_{out2,o} - t_{in2,o}) \frac{P_{q1,o}^*}{1 - \exp(-U_{1,o}^*)}$$

$$m_s^* = P_{q1}^* / (P_{q1,o}^* (1 + b(t_c - t_{c,o})))$$



$$t_c = t_{c,o} + \frac{-(t_{c,o} - t_{in1})(1 - \exp(-U_1^*)) + m_{st}^* \frac{P_{q1,o}^*}{P_{q1}^*} (t_{out1} - t_{in1})}{(1 - \exp(-U_1^*)) - b m_{st}^* \frac{P_{q1,o}^*}{P_{q1}^*} (t_{out1} - t_{in1})} \quad (5-27)$$

$$P_{qt}^* = m_{st}^* P_{q1,o}^* (1 + b(t_c - t_{c,o})) \quad (5-24)$$

$$t_1 = t_{in1} + (t_{out1} - t_{in1}) \frac{P_{qt}^*}{P_{q1}^*} \quad (5-28)$$

$$P_{qt}^* = P_{q1}^*$$

$$t_i = t_{out1}$$

$$P_e^* = P_{e,o}^* m_s^* (1 + a(t_c - t_{c,o}))$$

$$P_{eb}^* = P_{eb,o}^* m_s^* (1 + e(t_c - t_{c,o}))$$

$$P_{e1}^* = P_e^* - P_{eb}^*$$

$$P_{e2}^* = P_{e1}^* - P_{ec}^*$$

5.3 Presentation of Results

In this section we shall present a series of diagrams showing calculation results for the extended system model, depicted in fig. 5-1 and defined mathematically by the calculation procedure shown in table 5-1.

Fig. 5-6, containing four diagrams, is analogous to fig. 4-2 for the simple system model, without a peak heat-only boiler and with no hot consumption water production. From the top diagram, depicting the net electric output (as before, defined as the output from the steam cycle minus the pumping power for the DH system) we can make the following readings: At low heat loads (with the size of the turbine assumed in the figure, this means approximately -10°C), P_{e2}^* varies not very much from the picture shown for the simpler model; except at very low DH mass flowrates the output increases with growing heat load. But when full steam mass flow has been reached, i.e. when $m_s^* = m_{st}^*$ (m_{st}^* being defined as the maximum steam flowrate, in dimensionless form), then P_{e2}^* abruptly starts to decrease with increasing heat load. So long as $m_s^* < m_{st}^*$, the change in P_{e2}^* with increasing heat load is the result of two counteracting effects (cf. section 4.1): A higher heat load makes possible a larger steam flow through the turbine, but at the same time a higher back-pressure (resulting from a higher DH supply temperature) diminishes the enthalpy drop over the turbine. Normally, and particularly at lower heat loads, the first of these two effects is the strongest, so that a net increase in P_{e2}^* results. However, when maximum m_s^* is reached, only the negative of the two counteracting effects remains, thus the decrease in P_{e2}^* , when the peak boiler has gone into operation.

As in fig. 4-2, a line connecting points of maximum P_{e2}^* at each ambient temperature t_a has been drawn, not only in the top diagram of fig. 5-6, but also in the diagram below for supply and return temperatures. The resulting variations in temperatures and in DH mass flowrate m^* with ambient temperature is shown in the two diagrams to the right.

At low heat loads the introduction of hot water production is seen to have the effect of deflecting the supply temperature curve to a value some degrees above that of the (constant) hot water temperature t_{hw} . At medium heat loads (say, from -10°C to 5°C), the variations of supply and return temperatures, as well as that of the DH water flowrate, are similar to the

Table 5-2

Input values for parameters in table 5-1, valid for all results shown in diagrams in Chapter 5.

Constants:

t_s	=	0°C
t_h	=	20°C
t_{kw}	=	5°C
t_{hw}	=	55°C
n	=	1.3
x_l	=	0.1
x_r	=	1.0
x_{hw}	=	1.0
p_l	=	0.5
p_r	=	0.5
p_{hw}	=	0.5
y	=	0.5
a	=	-0.005
b	=	0
e	=	-0.001
i	=	-0.05

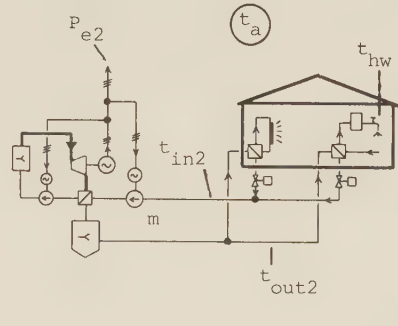
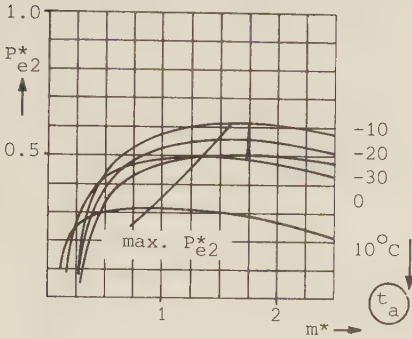
Values defining reference conditions:

$t_{a,o}$	=	0°C
$t_{1,o}$	=	50°C
$t_{out2,o}$	=	90°C
$t_{ina,o}$	=	60°C
$t_{inb,o}$	=	60°C
$U_{1,o}^*$	=	1.5
$P_{qIO,o}^*$	=	0.04
$P_{qIIO,o}^*$	=	0.04
$P_{qIII,o}^*$	=	0.04
$P_{ec,o}^*$	=	0.005
$P_{eb,o}^*$	=	0.005
$P_{e,o}^*$	=	0.5

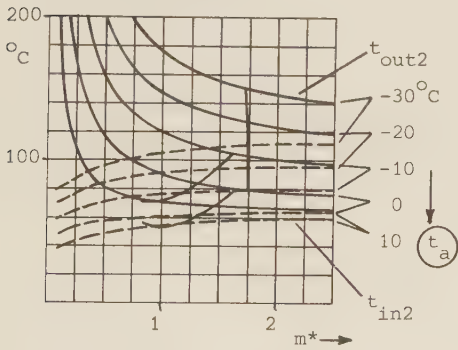
Fig. 5-6

Load performance of extended system model.
 DH operational modes to maximize net electric output P_{e2}^* at varying heat load.

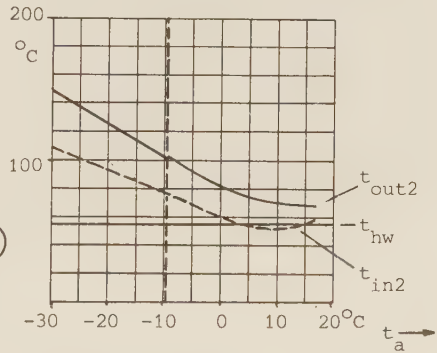
Net electric output:



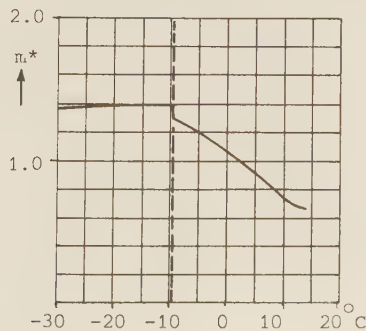
Water temperatures:



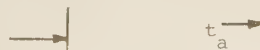
Water temperatures when $P_{e2}^* = \max$:



Water flowrate when $P_{e2}^* = \max$:



peak boiler
in operation



variation found for the simpler system model in fig. 4-2. At high loads, when the peak heat-only boiler is in operation, it is apparently no more optimal (in the sense of maximizing P_{e2}^*) to increase the mass flowrate with growing heat load.

At high heat loads, when $m_s^* = m_{st}^*$, the DH mass flowrate m^* is kept virtually constant to maximize P_{e2}^* . At lower loads, the marginal increase in turbine output P_e^* from each $^{\circ}\text{C}$ of lowering of condensing temperature grows with the size of the steam flow (which is related to the heat load), so that it is then optimal to increase m^* with the heat load. But when full steam load m_{st}^* is reached this kind of effect is no longer present, which explains the virtually constant value of m^* at high ambient temperatures.

From the diagram at the bottom of fig. 5-6 it is seen that when the peak boiler goes into operation there is an almost instantaneous, albeit small increase in optimal flowrate. This is a consequence of the fact that we have taken into account in the thermodynamical description of the DH system that the pumping power P_{ec} , added to the system, reduces the heat load P_{q1} for a given heat load P_{q2} . Since the steam flowrate to the turbine condenser is in turn determined by P_{q1} , this leads to a smaller turbine output P_e , so long as $m_s^* < m_{st}^*$. But when full steam flowrate has been reached, an increased P_{ec} will no longer reduce P_e , although it of course decreases the net output P_{e2} , a decrease which as usual must be weighed against the gain in lower condensing temperature, following a larger DH flowrate.

In fig. 5-6 the return temperature t_{in2} maximizing P_{e2}^* is seen to have a minimum, occurring at a relatively high ambient temperature. t_{in2} is the result of the mixing of the two partial flows with temperatures t_{ina} and t_{inb} , see fig. 5-7. In the numerical example for which results are shown, those two temperatures were selected to equal values at reference heat load, i.e. at $t_a = 0^{\circ}\text{C}$. At higher ambient temperatures, t_{inb} and the partial primary flowrate m_b^* both increase, while t_{ina} and m_a^* go down. The fact that the net effect of those opposing effects on the mixing temperature t_{in2} is a decrease, is explained by the fact that at $t_a = 0^{\circ}\text{C}$ flow m_a is much than flow m_b (although from fig. 5-7 their dimensionless values m_a^* and m_b^* are seen to be rather equal). At ambient temperatures above 0°C , m_b^* increases rapidly, which explains that with increasing t_a , t_{in2} reaches a mi-

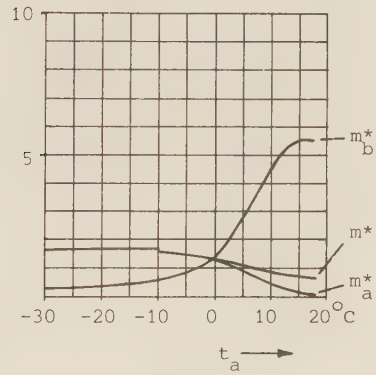
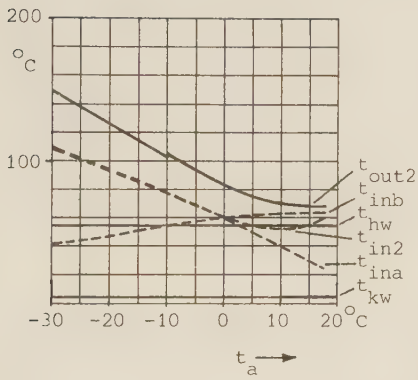
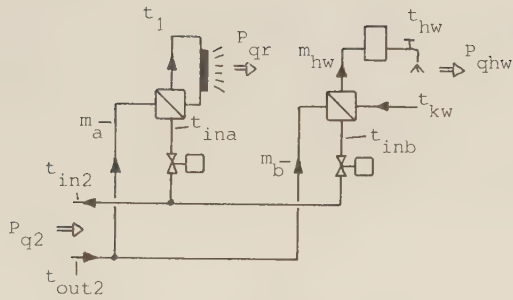


Fig. 5-7

Load performance of consumer's substation of extended system model with heat exchangers for radiator water heating and for building hot water services in parallel.

DH network operated to maximize net electric output P_{e2} at all load conditions.

nimum, and later increases. In the limiting case of vanishing space heating requirement, t_{in2} becomes equal to t_{inb} .

The kind of DH supply and return temperature variation, as well as the variation of the DH mass flowrate, depicted in fig. 5-6, are in their general character in accordance with practice, as reported both by statistics from the UNICHAL organization [5-4], and by single DH companies (e.g. [5-3] and [5-5]). There seems to be a general tendency that in Nordic countries, with their colder winters, supply temperatures are kept constant or virtually constant down to lower ambient temperatures than what is practiced in Central Europe.

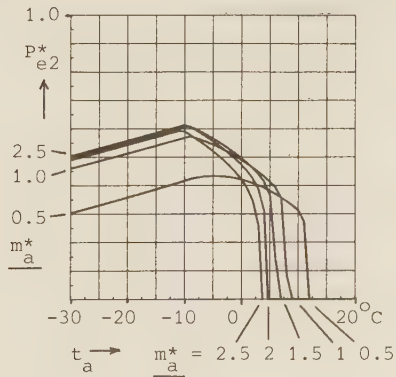
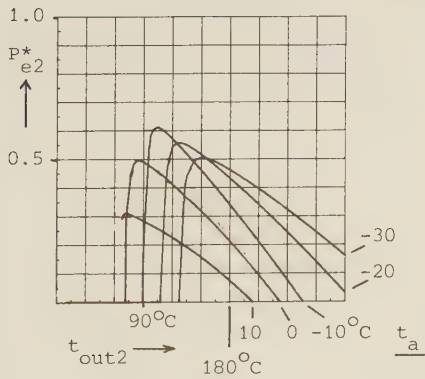
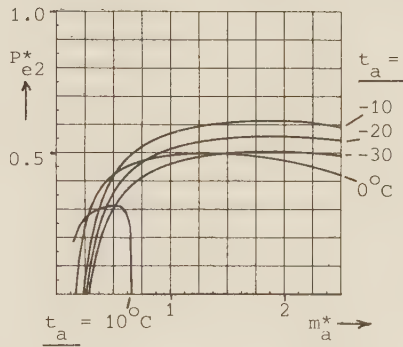
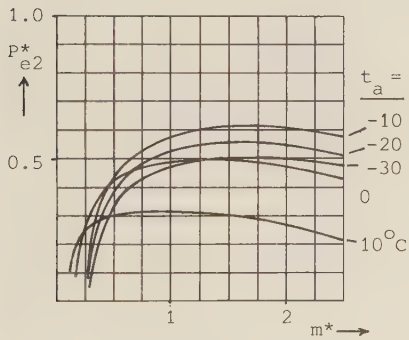
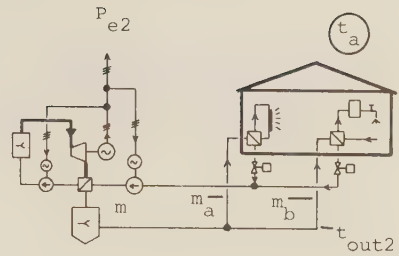
Further insight into the operational characteristics of our extended model is provided by fig. 5-8, which is similar in its composition to fig. 4-5 valid for the simpler model. However, in fig. 5-8 two diagrams with dimensionless DH mass flowrates on the abscissa are given. The lower one, with the total dimensionless mass flowrate m^* on the abscissa is the most direct counterpart to the diagram at the top left of fig. 4-5. In the calculation procedure of table 5-1 m_a^* , not m^* , has been selected as the variable characterizing the single degree of freedom of the DH system, and at computer calculations successive numerical values are assigned to m_a^* . Thus, the diagram at the top of fig. 5-8 may be informative for the reader wishing to penetrate the calculation technique adopted here.

An observation which can be made from the bottom left-hand diagram of fig. 5-8 is that the asymmetry of the curves becomes more pronounced with increasing ambient temperature. As has been pointed out already when discussing similar curves for the simplest system model, the asymmetry from a practical control point of view leads to a recommendation to set the supply temperature at a value above that which gives absolute maximum for P_{e2}^* . It is readily seen that irrespective of what criterion is applied when selecting DH supply temperature, t_{out2} must at all loads be above the hot consumption water temperature t_{hw} . With space heating only, as in the simple system model, considerably lower supply temperatures have been found to be optimal at high ambient temperatures. This means that in the summer season provision for hot water forces the supply temperature to take a value giving a DH heat loss which may be high in comparison with the total amount of heat delivered by the system. If in the summer season electric energy is rated

Fig. 5-8

Four ways of presenting results
for net electric output from
extended system model.

Maximum steam flowrate through turbine $m_{st}^* = 1.4$.



economically no higher than in parity with heat from fossil fuels, the condition of maximum net electric output P_{e2}^* no longer appears to be a reasonable criterion for selecting an operational mode for the DH system. Instead, the supply temperature should then be set as low as possible to minimize heat losses, but high enough to ensure proper system control at all types of disturbances which may occur in the system.

The envelope curve of the diagram to the right at the bottom of fig. 5-8 shows maximum P_{e2}^* as a function of t_a , except at high ambient temperatures where values of m_a^* lower than 0.5 would have to be added to extend the envelope. Alternatively, the total mass flowrate m^* could have been selected as curve parameter. The decline in P_{e2}^* with decreasing t_a at values of $m_s^* = m_{st}^*$ is seen to be close to a linear relationship.

At the top of fig. 5-9 envelopes for P_{e2}^* have been drawn for various values of the limit steam flowrate m_{st}^* , together with diagrams of other important system parameters. It can be seen that the nearly straight lines of P_{e2}^* -envelopes display an increasing slope with increasing limit steam flow m_{st}^* . A practical consequence is this: When the maximum net electric output is examined at a given, low ambient temperature for different sizes of the turbine plant, it is true that the output decreases with decreased turbine size, but the loss in P_{e2}^* is not proportional to the reduction in turbine size. This observation may be explained by the fact that when the back-pressure turbine covers a smaller part of the total heat demand, the steam condensing temperature t_c (shown in the diagram below that for P_{e2}^*) is lowered, since it is related primarily to t_i , the intermediate temperature between supply and return temperatures, as shown in fig. 5-5. The lower steam condensing temperature is beneficial for the size of the electric output.

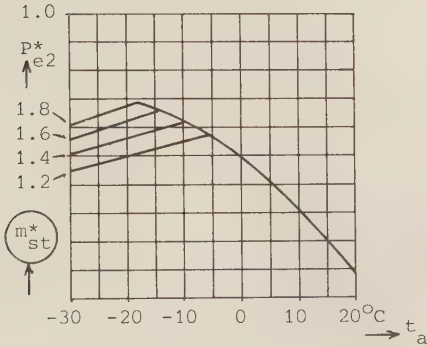
In this section several diagrams have been shown which are counterparts of similar diagrams from section 4.1, valid for the simpler system model without a peak heat-only boiler and with no consumption hot water production. The construction of load duration curves for net electric output in fig. 5-10, which resembles the construction shown in fig. 4-7, completes this series of graphical parallels. These parallels have shown, as was claimed in section 1.4 (Method of Analysis) that the simplest model should not be regarded as merely a more primitive representation of reality. By drawing

Fig. 5-9

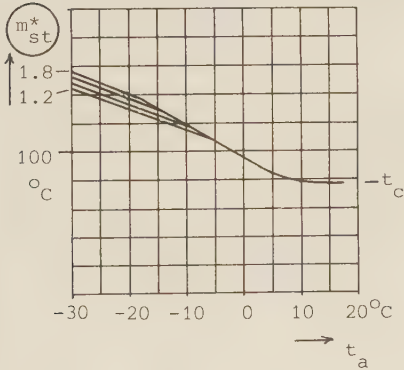
Performance of extended system model.

Effects of varying size of maximum steam flow-rate m_{st}^* on operational modes to maximize net electric output P_{e2}^* .

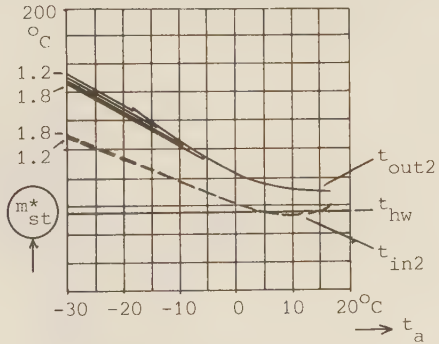
Net electric output:



Steam condensing temperature:



DH water temperatures:



DH water flowrate:

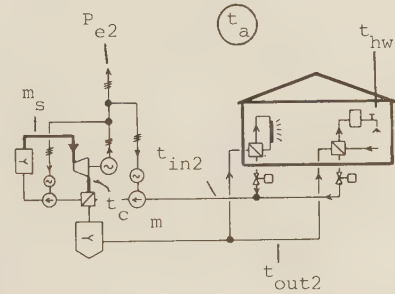
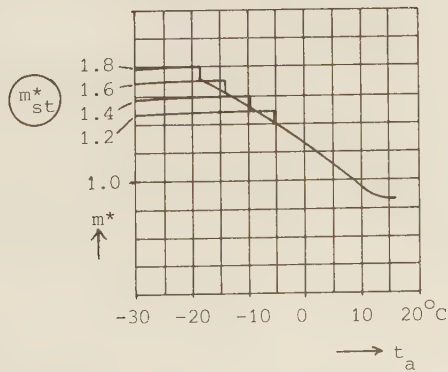
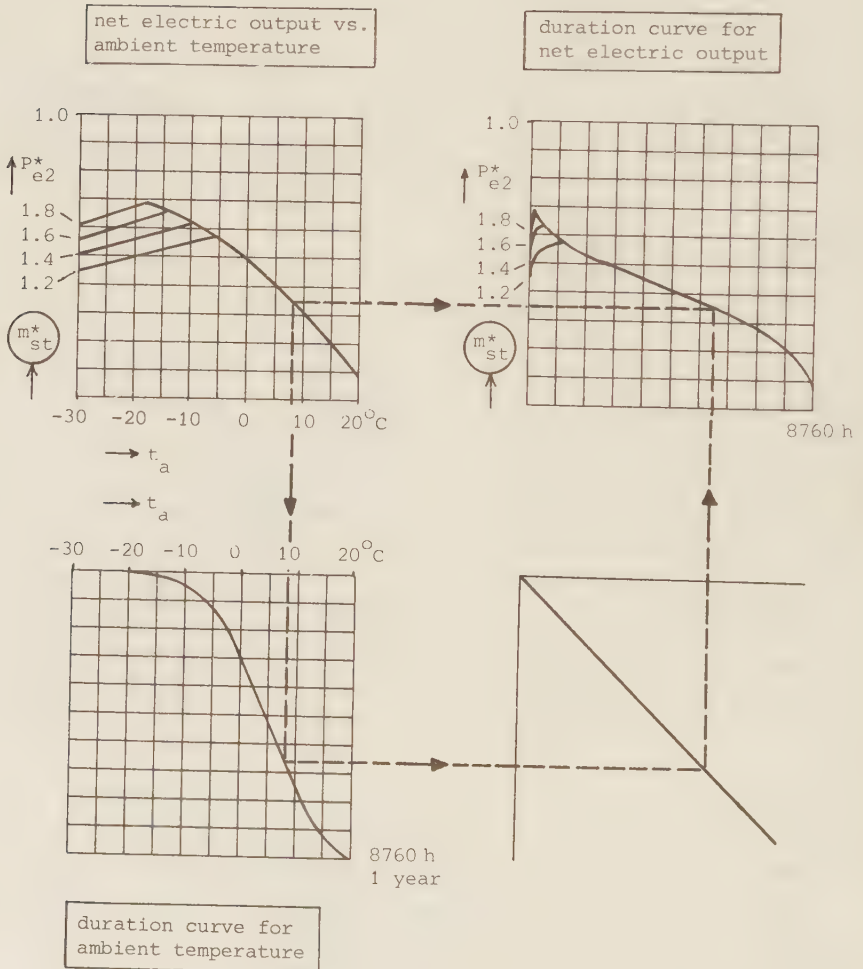


Fig. 5-10 Construction of load duration curves for net electric output from extended system model.

Curve parameter: maximum steam flowrate in back-pressure plant.



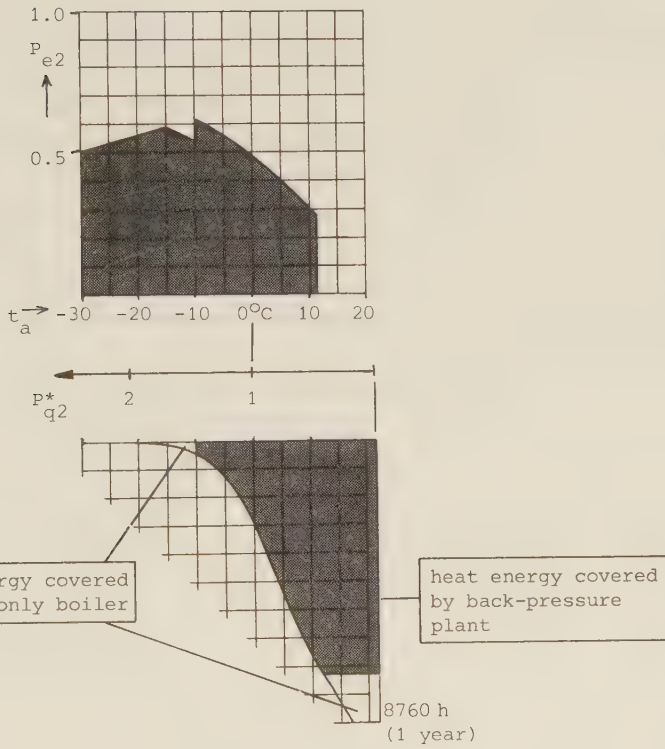
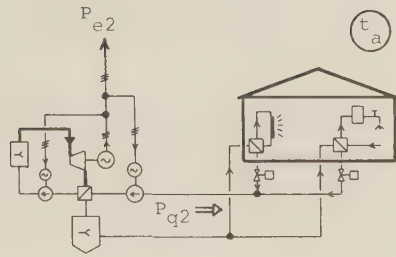


Fig. 5-11 Yearly heat energy demand covered by back-pressure plant and peak heat-only boiler.

Minimum ratings for both types of generating plant.

upon observations made from the simplest model, a deeper understanding for the more sophisticated model is reached.

In section 1.1 we drew attention to the observation that the system load of big electric networks tends to increase with a lowering of the ambient temperature t_a . The results derived in this section for the net electric output P_{e2} show that by selecting the DH operational strategy appropriately it is possible to match partly this load variation at higher ambient temperatures. But at some DH heat load the peak boiler must into operation, and from then on a further lowering of t_a will give a variation of the output which is opposite to the variation in electric system load, even if the DH operational strategy is selected optimally.

Fig. 5-11 finally shows the effect on the envelope for net electric output of minimum ratings for the two boilers in the system. Here, the back-pressure plant is shut down for a period in the summer, so that the electric output vanishes. The minimum rating of the peak heat-only boiler is seen to give a jagged form of the P_{e2}^* at its peak value, when the heat-only boiler goes into operation.

6. NOTES ON NON-STATIONARY OPERATION

The numerical results derived in the preceding chapters all apply to static operational conditions. With some caution they may be interpreted as mean values for time-dependent variations. However, genuinely dynamic effects cannot be expected to be covered by equations derived for static load conditions. The purpose of this chapter is to provide some insight into the nature of non-stationary operation, on a qualitative basis, and without any ambition to present a complete picture.

We shall be concerned with such non-stationary effects which are associated with a temporary storage of heat in various parts of the system and with time lags between heat generation and consumption. Such phenomena usually take place over one or several hours, since both transportation times for water in a DH system and time constants for building construction materials are of this order of magnitude.

Dynamic effects associated with the presence of pressure variations in the water are of a much faster nature, due to the rapid propagation of sound in water. As was pointed out in section 4.2, such phenomena may be important for determining technological limits for the operation of DH systems, but they will not be dealt with below. Another important group of non-stationary phenomena excluded in this chapter is dynamic control problems. An example of an important question of this kind is to what extent a DH system is capable of following rapid changes in heat load. The more extended the system is, the greater is the average time lag between heat generation and consumption, so that the greater is the risk that water arriving at a consumer has an inappropriate supply temperature.

One common source for non-stationary effects in a DH system is the fluctuation of the ambient temperature during the day. Fig. 6-1 illustrates an account for such fluctuations in a purely static system consideration. In the bottom left diagram are depicted two typical variation patterns for heat loads, as given by variations in ambient temperature. As pointed out by e.g. Zängl [1-33], amplitudes in ambient temperatures during a 24-hour cycle are normally greatest in the summer months. The minimum temperature is generally reached around sunrise.

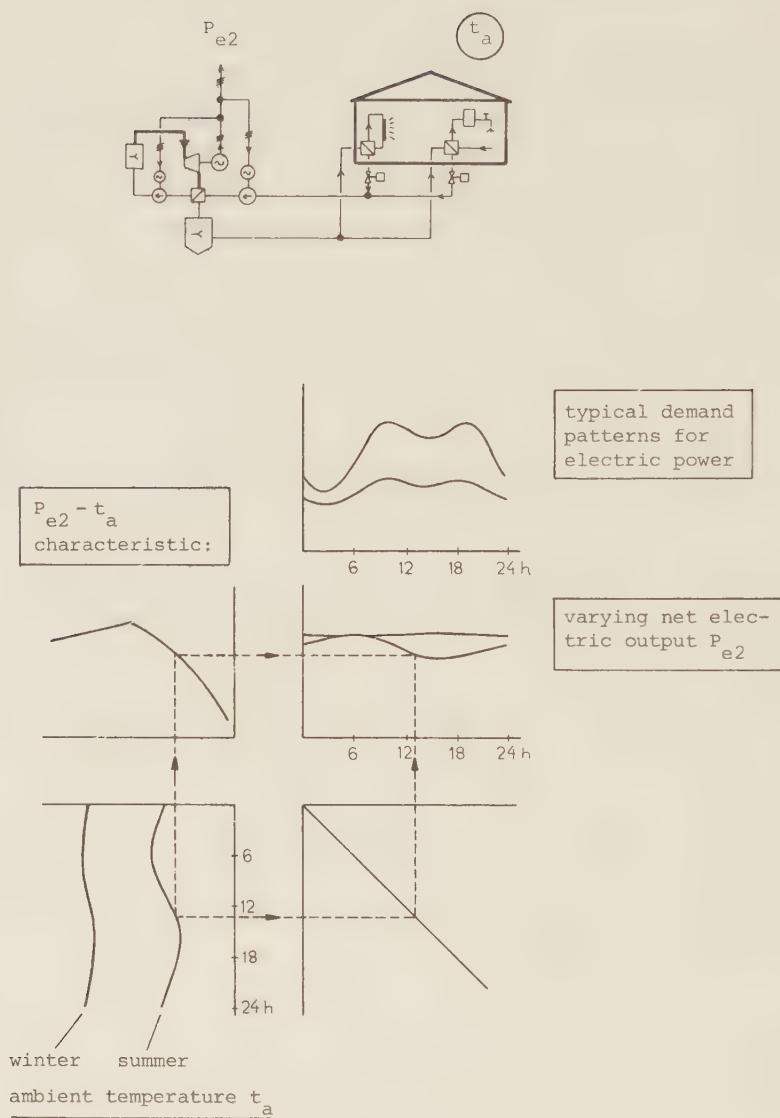


Fig. 6-1

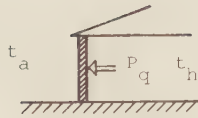
Construction of curves for variation of net electric output during a 24-hour cycle. Temporary heat storage in building walls and in DH system not taken into consideration.

In the figure the two heat load variations have been combined with a $P_{e2}^* - t_a$ curve for a back-pressure plant in series with a peak heat-only boiler, as analysed in Chapter 5. The resulting variations in net electric output over a 24-hour period are seen to be quite different for the two load cases. In summer the output is found to vary much more than in winter, not only because of the greater variation in ambient temperature, but also due to the greater slope of the $P_{e2}^* - t_a$ curve at lower heat loads. The two P_{e2}^* vs. time curves are seen to be in opposite phases, caused by different signs of slopes of the $P_{e2}^* - t_a$ curve. The resulting time variations for net electric output may be compared with typical patterns for variation of loads in electric supply networks, as depicted at the top of fig. 6-1. Roughly speaking, in each of the two cases considered, one of the two demand peaks is seen to coincide with peaks in output.

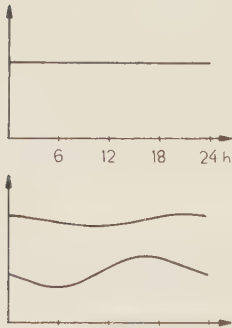
A construction of the type shown in fig. 6-1 may be valid in some cases. However, the fact that it does not account for non-stationary effects may give quite misleading results in other cases. As we shall see below, heat storage in building materials may cause the variation of the heat load on the DH network to differ significantly from the variation of the heat load, given by the ambient temperature. Also, the heat load at the generating plant may vary quite differently from the heat load at the consumer. Thus, the resulting net electric output may differ greatly from the pattern shown in the figure.

Fig. 6-2 illustrates effects of heat storage in building construction materials, as estimated from results reported in references ([1-33] and [3-3]). In all cases the same variation of ambient temperature has been presupposed, while the room temperature has been assumed to change according to different patterns. Resulting heat loads to be supplied by the building heating system are seen to differ greatly among the four cases.

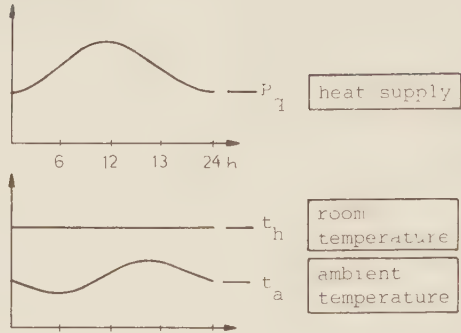
In case A a constant heat flow P_q is supplied to the building. The result is that there is some variation in the room temperature t_h . This variation is, however, smaller than the variation in the ambient temperature t_a , i.e. the walls dampen temperature variations. It can be seen that there is a phase shift, i.e. maximum room temperature occurs some hours after maximum ambient temperature has been reached. Increased wall thickness will result in greater damping and in a larger phase shift.



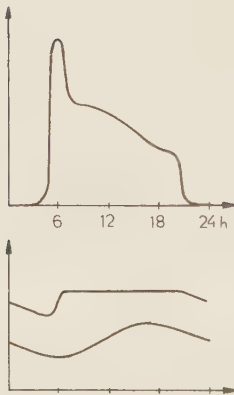
Case A:



Case B:



Case C:



Case D:

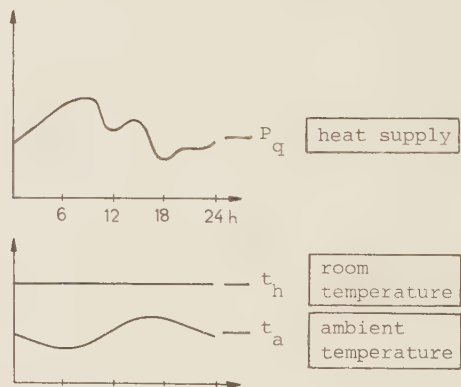


Fig. 6-2

Four cases of temporary heat storage in building walls. Different variations in building heat supply P_q for identical variations in ambient temperature t_a .

If the room temperature is kept constant in some way, as indicated in case B, considerable variation is found in the size of the heat supply. It may seem surprising that it takes a great variation in heat flow to eliminate a small variation in room temperature. The explanation can be found in the theory of harmonic waves, applied to the non-stationary heat conduction in the walls [3-3]. The wavelength of the temperature profile in the walls is not great compared to the wall thickness, so that the heat flow on the inside surface may at times reach high values.

If maximum thermal comfort for the people occupying the rooms is given by a certain temperature level, the temperature swing in case A leads to a greater integrated heat supply over a 24-hour period, when the room temperature is never allowed to fall below the minimum level. A constant room temperature may be achieved at the cost of a more sophisticated control system for the radiators. In principle thermostats, measuring the indoor temperature and regulating the radiator water flows, could secure a constant room temperature. However, the fact that variations in room temperature are small would make such a control sensitive to disturbances, so that the control should incorporate a direct measurement of the ambient temperature, from which necessary adjustments in heat supply can be made [6-1].

Intermittent occupance schedules for buildings may permit temporary room temperature setbacks, usually during some hours at night. By controlling heat supply accordingly, savings in integrated heat supply over a 24-hour cycle can be achieved. Case C illustrates such a pattern. It is characteristic that here there is a peak in the heat demand in the early morning hours to cover, not only a rise in the room temperature, but also heat to be stored in the walls. In the winter it may not be acceptable to have the heat supply shut off all night, since this may cause water condensation on walls with a low surface temperature. In such cases it may be necessary to have a moderate heat supply in the latter part of the night to keep surface temperatures at a constant level.

The amount of heat stored in building walls, gradually given up to the room at night, and to be supplied again in the morning, reduces the savings in heat supply gained from a lower room temperature at night. When the wall insulation material is placed on the inside of the walls, the pulsating heat flow is reduced, and savings increase accordingly [3-3].

When buildings are heated by domestic boilers or by electric resistance heating, savings gained from nightly setbacks of room temperature may be evaluated fairly easily. In cases where the heat supply is instead served by a DH system, it must be estimated how the variation in building heat demand affects heat generating plants. Bode [6-2] has pointed out that when the heat demand for a DH is covered by several heat-only boilers, losses due to starting of boilers or due to increased fuel supply to keep a peak-ing boiler warm before going into service, may reduce or even cancel savings achieved by reducing room temperatures at night.

In the case of a DH system served by a back-pressure plant, reductions in building heat demand by reducing room temperatures at night may be utilized for reducing primary energy consumption, as with heat boilers. In addition, variations in electric output with time must be considered when evaluating this type of heating method. So long as transportation times for water from generation to consumption are small, and maximum steam flowrate through the turbine has not been reached, it can be said that a reduction in the heat demand at night is favourable; it will lead to a reduced electric output when the demand for electricity is also generally low. However, the peak in heat demand for buildings in the early morning hours occurs before the demand for electricity has gone up significantly. Also the time lag of the DH system must be taken into consideration; it will cause the peak in electric output to occur even earlier in the morning. In cases where maximum steam flowrate through the turbine has been reached, peaks in heat demand will cause drops in electric output, due to a higher supply temperature determining the turbine back-pressure level. In conclusion, it is difficult to make general statements about the economic value of intermittent building heating schedules in the case where DH is based on a back-pressure plant.

In the final case of fig. 6-2, case D, effects of building gains of solar heat are shown. Around noon, at maximum altitude of the sun, there is a peak in heat inflow through windows. Later in the day a new peak is found, attributable to solar radiation on building external surfaces, causing a delayed reduction in the heat loss from the building interior.

While fig. 6-2 referred to temporary heat storage in building walls, fig. 6-3 shows an example of heat storage in the DH network, in the idealized case of a single heat consumer. The heat flow supplied to the consumer is assumed

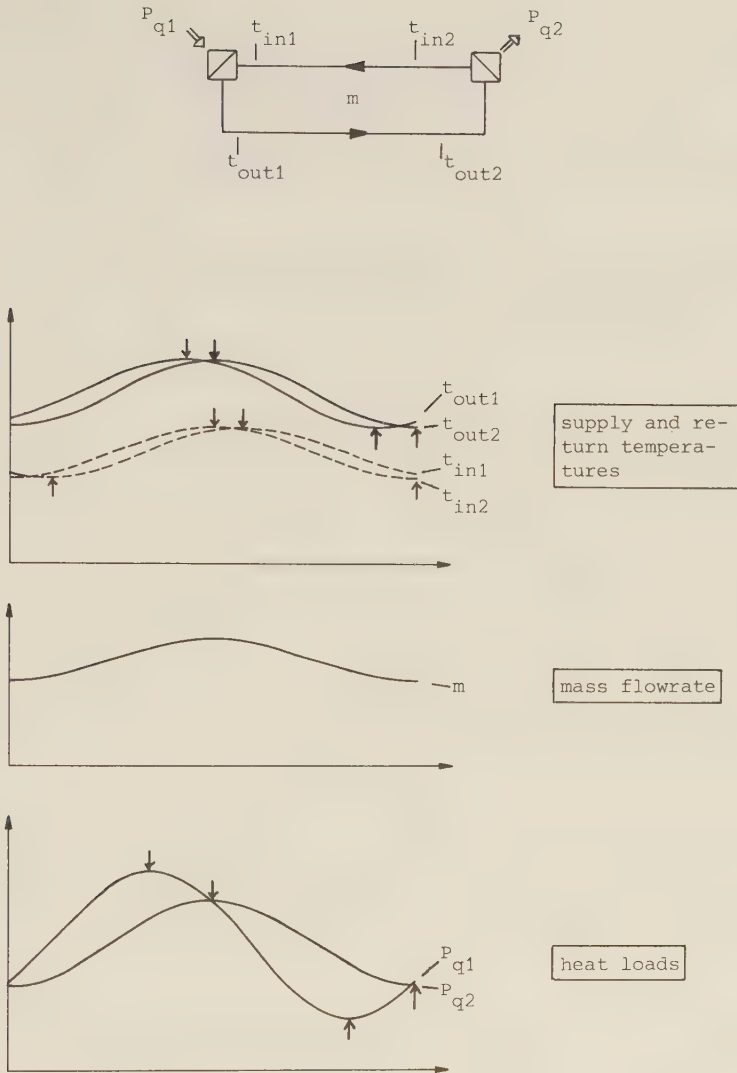
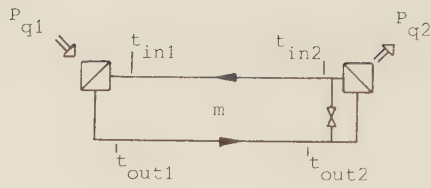


Fig. 6-3

Example of time-varying heat input P_{q1} to satisfy a harmonically varying heat demand P_{q2} in a DH system with a time lag.



Heat storage in
supply pipe:

Heat storage in
return pipe:

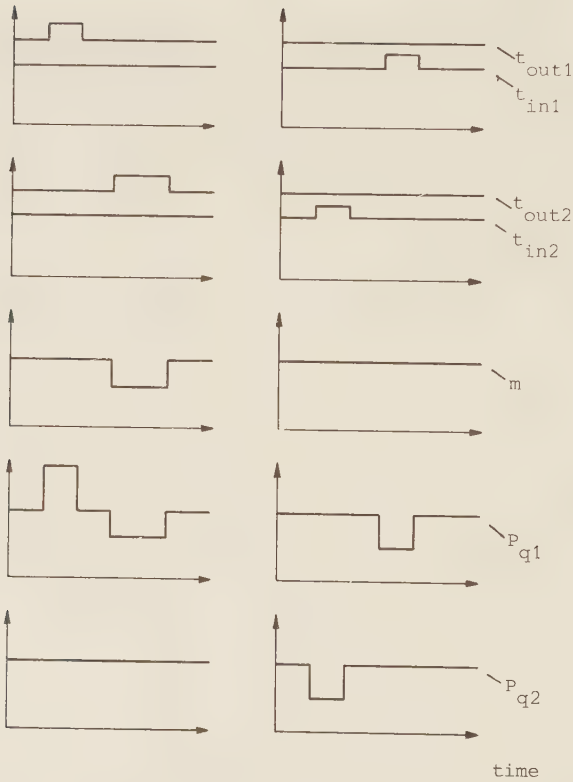


Fig. 6-4 Two idealized cases of temporary heat storage in DH pipelines.

to vary sinusoidally, which as we have seen above may be needed in some cases of dynamic building heat load conditions. Both the supply temperature t_{out2} at the consumer and the DH water flowrate m^* are assumed to follow variations in P_{q2}^* , as in typical static operation. The return temperature t_{in2} will vary accordingly as indicated by the figure. By taking into account water flow time lag which itself varies inversely with the size of the flowrate, variations in the temperatures t_{out1} and t_{in1} at heat generation can be calculated, assuming for simplicity that DH heat and pumping losses are negligible. These two curves, in combination with the variation of the flowrate, give the curve for the variation of the heat input P_{q1}^* to the system.

P_{q1}^* is found here to vary with a greater amplitude than P_{q2}^* , and with a phase shift which is greater than the time lag of the water transported from generation to consumption. In those time intervals where P_{q1}^* is in excess of P_{q2}^* , a storage of heat in the system takes place, and when P_{q1}^* is smaller than P_{q2}^* , heat is tapped from the DH system. It is remarkable that the amplitude in the variation of heat flow is magnified so much from P_{q2}^* to P_{q1}^* , even with the rather small time lag assumed in the example of the figure. It may in unfavourable cases even occur that the heat input temporarily vanishes, which can be considered as unacceptable.

Although time lags in DH systems may cause problems of operation, temporary heat storage in DH systems may also be beneficial when utilized strategically, as has been investigated by Suttor [1-41]. Fig. 6-4 shows two examples of this, the first involving heat storage in the supply pipe, the second storage in the return pipe.

In the first case the supply temperature is suddenly raised at a constant water flowrate, so that there is an increase in the heat input to the system. At first there is no change in the supply temperature at the consumer, i.e. the heat output from the system is kept unchanged, and heat is stored in the system. At a later time the supply temperature at heat generation goes back to its initial level. At a still later time the temperature increase in the supply line reaches the consumer. If the heat demand of the consumer is to be kept unchanged, the water flowrate must be reduced, causing a decrease of the heat input at heat generation. At the point of time when the supply temperature at the consumer returns to its initial level, the amount of heat

stored in the system at the beginning of the time interval depicted has been removed again. By operating a system in the way described here, it is possible to increase momentarily the heat flow to the DH system at one time, accepting a reduction in the heat input at a later time. In this way it is possible to generate peak electricity from a back-pressure plant by utilizing the DH system as a storage capacity.

For the sake of simplicity variations in supply temperature at heat generation in fig. 6-4 have been assumed to be transmitted unaltered to the consumer, except for changes in timerates, caused by variations in water flow-rate. In real pipelines there will be some axial mixing, since water particles near the inner surface of the pipe flow more slowly than those in the core of the flow. Such mixing will soon 'round off' sharp edges in temperature profiles, but profiles with moderate slopes will probably remain fairly constant when propagating along the pipe.

In the second case of fig. 6-4 heat is stored temporarily in the return line by raising the return temperature at constant water flowrate in the system. This can be done by opening for a by-pass flow at the consumer. When the raised return temperature reaches the heat generating plant, the heat input is assumed to be reduced, keeping the supply temperature constant at a constant water flowrate. By adopting this method of heat storage in the return pipe it is possible to postpone a reduction in heat generation, following a sudden reduction in heat demand. This can be favourable in cases with a back-pressure plant in peak operational mode at the time of reduction in heat demand.

Halzl et.al. [6-3] investigated for CHP plants in practical operation the possibilities of generating temporary peaking electric output by utilizing DH systems in non-stationary operation. They found that increases in output by as much as to a doubling are possible without impairing stable heat supply to buildings. The investigation drew attention to effects of variations in demand for hot consumption water. In the mornings this demand displays a peak which can be utilized for generating a surplus of electric output from back-pressure plants.

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8. SUMMARY

The thesis presents a thermodynamic analysis of district heating (DH) systems based on the circulation of hot pressurized water, utilizing heat from a back-pressure steam power plant. Performance characteristics for simple models of such systems are calculated. A main variable selected for analysis is the net electric output, defined as the turbo-alternator output reduced by the power consumption of the DH pump maintaining the hot water flow. The size of this variable is calculated for various values for DH design parameters and for various load conditions; these include both changes in heat load and in the DH operation strategy, given by the water mass flow-rate and by the water supply temperature. The numerical results derived are shown in diagrams, utilizing alternative graphical techniques to demonstrate various aspects of system performance.

Calculations are made for two system models. In both models the DH network is simplified to consist of a single pair of supply and return pipes, serving a single (big) heat consumer at a distance from the plant. The steam plants are based on a simple Clausius-Rankine cycle with no regenerative feedwater heating. The first model supposes space heating only for the building, and relies on the back-pressure plant alone to cover the heat load. In the second and more realistic model, hot water services are provided for as well, and a peak heat-only boiler is connected in series with the back-pressure plant. Although both models are highly simplified in some respects, they are shown to possess a number of features pertaining to more complicated systems encountered in practice. Care is taken in the analytical description of both models to represent DH heat and pressure losses in a thermodynamically correct way; changes in water temperatures along the supply and return pipes are accounted for, and the effect of DH losses on the size of the heat load covered by the generating plant is included.

In the first section of Chapter 1 (Introduction) the theme of the thesis is presented and the aim of the study is formulated. Attention is drawn to some current trends in the development of the electric power system load in Sweden and elsewhere; performance characteristics of back-pressure plants should be discussed with those trends in mind. In section 1.2 the present investigation is related to previous monographs and to DH literature in general. It is argued that a common basis of knowledge is lacking, which is a reason for

a number of elaborations on comparatively elementary points in the text. In section 1.3 the methodology adopted in the thesis is discussed, making clear what is meant by characterizing the method as a thermodynamic systems analysis based on simple models. Section 1.4 summarizes the mathematical formalism employed in the work; both reference system modes and dimensionless variables are utilized extensively in the analysis.

In Chapter 2 (Model Formulation) the two models investigated in the thesis are presented and discussed. Sections 2.2 and 2.3 contrast model solutions for consumer connections and for steam plant layout, respectively, with more complicated variants. In section 2.4 the model assumption of a single pair of supply and return pipes and of a single heat consumer is discussed. The presentation made in this chapter makes explicit various model limitations, thereby facilitating possible future formulation of more refined models.

In Chapter 3 (Characteristics of System Elements) analytical expressions are derived for the performance of each of the main system elements of the simple model. The discussion in section 3.1 on radiator building heating is mainly a recapitulation of well-known theory. In section 3.2 an attempt is made to give a rigorous treatment of DH heat and pressure losses. For model calculations, the heat flow situation round pipelines is found to be suitably represented by three superimposed, linear heat flow components. The DH pumping power, derived from pressure losses, is divided into components related to the supply and return pipes, and to the steam plant condenser as well as the consumer's heat exchanger. Section 3.3 gives equations for the performance of the two heat exchangers. Special attention is given to effects of very large pressure drops in the heat exchangers, in theory possible at system operation with great water mass flowrates. In section 3.4 performance equations for the steam cycle are given, linearizing the influence of varying steam condensing temperature. Possible variation in turbine expansion efficiency is discussed. Section 3.5 finally deals with the performance of the power station boiler.

In Chapter 4 (Performance of Simple System Model) the simplest of the two models is investigated. Section 4.1 deals with load performance, i.e. various operational modes for given system dimensions, while in section 4.3 consequences of varying DH design parameters are calculated. Design parameters investigated are: pipeline length and diameter, pipeline insulation, and

heat exchanger dimensions. Parameters characterizing heat exchangers are: dimensionless heat transfer surface and a dimensionless 'heat exchanger length', determining the flow resistance of the apparatus. When varying system design parameters, the DH operational mode is changed simultaneously; thus it is possible from the results to read how the operational mode should be varied with system dimensions to ensure optimal performance, in particular how maximum net electric output is ensured. In performance calculations the operational performance is allowed to change without restrictions; in section 4.2 the impact of various technological restraints, such as maximum water velocity and minimum water pressure to avoid boiling, are studied on a qualitative basis, utilizing graphical techniques.

Chapter 5 (Load Performance of Extended Model) presents results for the more refined of the two system models, which includes building hot water services and a peak heat-only boiler. At lower heat loads, to ensure maximum electrical output, only the back-pressure plant is operated. At some heat load maximum steam flowrate through the turbine is reached, so that at greater heat loads the heat-only boiler must be operated as well. Operational modes to maximize net electric output at various heat loads are identified. Diagrams are given, both for this output itself, and for DH supply temperature and water flowrate as a function of the ambient temperature. The DH operational strategy found is seen to be in accordance with common plant practice.

Chapter 6 (Notes on Non-Stationary Operation) gives a short and qualitative discussion on effects of temporary heat storage in various parts of the system. In practice phenomena of this kind will often play an important role along with static system characteristics, which is the main subject of the thesis.

